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CENTRIFUGAL FAN CALCULATIONS BY THE "SPECIFIC SPEED" METHOD.

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The purpose of this paper is to develop and illustrate a new method of solving problems arising in the selection, installation, and operation of centrifugal fans. A considerable amount of the troubles experienced with fan installations result from the misapplication of fans to conditions for which they are not suited. The task of selecting a fan is divided into two parts: first, knowing the requirements as to quantity of air and head, and second, determining the proper type, size and speed of fan to best meet these requirements. This paper is concerned mainly with the second part of this task, and the methods developed will assist in selecting the proper type of fan to be used, and will determine the correct size, speed and power to meet the requirements of a given installation.

The essential element of what the writer chooses to call the "specific speed method" of performing fan calculations con-

sists in expressing the diameter and speed of fans as functions of the quantity and head. The value of this method of treating the subject rests on the fact that in nearly all fan problems the quantity and head are known factors, and therefore by considering these as the independent variables, a simple and direct solution of problems may be obtained.

There will also be presented a method of classifying and comparing fans which is based on the duty the fans are able to perform, and is independent of any of the details of design. Evidently this is a logical basis for classification, as fans with different structural features may give practically the same performance, and fans of apparently the same design may vary in performance, due, for instance, to small differences in blade design. The fan user is interested primarily in the performance of the fan, and secondarily in the details of design.

LAWS OF FAN PERFORMANCE.

The formulae derived herein are based on two groups of laws which are well known to fan manufacturers. These laws are subject to theoretical demonstration, and have been proven to be substantially correct by numerous tests of fans of various types and sizes. The first group of laws pertains to a given fan operating on a system with a fixed external restriction. If the speed (rev. per min.) of the fan be changed, without changing any of the external restriction against which the fan operates, the following laws are effective:

(*a*) The volume of air discharged by the fan will vary as the speed.

(*b*) The pressure maintained by the fan will vary as the square of the speed.

(*c*) The power required by the fan will vary as the cube of the speed.

(*d*) The efficiency of the fan will remain constant.

A second group of laws may be stated which will cover the change in size of a fan of a given design. In this case consider a fan of a certain design and size, operating at a given speed, and on an external system with a given restriction.

The external restriction may be replaced by what is known as an "equivalent orifice," that is, an orifice at the outlet or inlet of the fan, of such a size that the performance of the fan is the same as when operating against the actual system. There may be constructed a fan of the same design as the one considered, but of different size, such that all dimensions of the new fan, including the "equivalent orifice," are in some exact proportion to the corresponding dimensions of the given fan. This fan may be designated as a "geometrically similar fan." If the geometrically similar fan be run at the same speed (rev. per min.) as the original fan, the following laws connect the performance of the fans:

(*e*) The volume discharged will vary as the cube of the ratio of the diameters of the fans.

(*f*) The pressure maintained will vary as the square of the ratio of the diameters of the fans.

(*g*) The power required will vary as the fifth power of the ratio of the diameters of the fans.

(*h*) The efficiency will remain constant.

It is not the purpose of this paper to prove the foregoing laws,* but to use them immediately in deriving the fundamental formulae of this study.

DERIVATION OF FORMULAE.

D = Maximum diameter at tip of blade, inches.
N = Revolutions per minute, or speed.
h = Static head, inches of water, at standard air density.*
Q = Discharge, thousands of cubic feet per minute.
P = Power required—horsepower.
V = Tip speed, feet per minute.
e = Efficiency.

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*For a theoretical proof of these laws and the effect of air density see paper, "Blower Performance," by Wm. L. De Baufre, JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, February, 1916. Also see the following books: "The Fan," by Chas. H. Innis, page 56; "Fan Engineering," Buffalo Forge Co., page 179; "Mechanical Engineer's Hand book," Marks. Published by McGraw Hill, 1916; section on fans by E. B. Williams, page 1545, bottom.

Laws (*a*), (*b*), (*c*) and (*d*) may be expressed as follows:

$$Q = C_1 N \quad . \quad . \quad .

may call C_{10} the "specific speed" of the fan, and designate it by the symbol N_s, so that

$$N = N_s \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and

$$N_s = N \frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

It should be borne in mind at this point that equations (11) and (12) make no mention of the diameter of the fan required to meet these conditions. The variation of diameter, as effected by quantity and head, for constant efficiency, and independent of speed, may be determined by eliminating N from equations (7) and (8) as follows:

$$\text{from (7)}; \quad N = \frac{Q}{C_7 D^3}; \qquad N^2 = \frac{Q^2}{C_7^2 D^6}$$

$$\text{from (8)}; \quad N^2 = \frac{h}{C_8 D^2}$$

equating and solving for D

$$\frac{Q^2}{C_7^2 D^6} = \frac{h}{C_8 D^2}; \qquad D^4 = \frac{C_8}{C_7} \frac{Q^2}{h}$$

$$D = C_{13} \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

For the special case in which $Q = 1$ and $h = 1$, $D = C_{13}$, or C_{13} is the diameter of a fan required to produce one inch static pressure, and one thousand cubic feet of air per minute. We may call C_{13} the "specific diameter" of the fan, and designate it by D_s; then

$$D = D_s \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

and

$$D_s = D \frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

DEFINITIONS.

We may now define the "specific fan" of a certain type of fan as a fan which is geometrically similar to a given fan

of the type, and has such a diameter, D_s, and runs at such a speed, N_s, as to deliver one thousand cubic feet of air per minute at one inch static head at any given efficiency.

The "specific speed," N_s, of any fan, operating at a given efficiency, is the speed of a geometrically similar fan which is of such a diameter that it will produce one thousand cubic feet of air per minute at one inch static head at the same efficiency.

The "specific diameter," D_s, of any fan, operating at a given efficiency, is the diameter of a geometrically similar fan which is running at such a speed that it will produce one thousand cubic feet of air per minute at one inch static pressure, at the same efficiency.

SIGNIFICANCE AND USE OF N_s AND D_s.

If for any fan, operating at a certain efficiency, the head, quantity, speed and diameter are known, the specific speed of the fan at that efficiency may be determined from equation (12),

$$N_s = N \frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}}$$

and the specific diameter at that efficiency may be determined from equation (15),

$$D_s = D \frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}}.$$

Conversely, if the specific speed and specific diameter of a certain fan or type of fan be known at any efficiency, the actual speed and diameter of a geometrically similar fan required to produce any desired quantity and head, at the same efficiency, may be determined from the formulae (11) and (14);

$$N = N_s \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}} \text{ and } D = D_s \frac{Q^{\frac{1}{2}}}{h^{\frac{1}{4}}}.$$

SPECIFIC POWER.

The power required by the specific fan, at any given efficiency is best obtained from the expression for air horsepower;

$$\text{Air H.P.} = \frac{\text{Quantity} \times \text{head} \times 5.2}{33{,}000}$$

in which quantity is expressed in cubic feet per minute, head in inches of water, and 5.2 is the pounds per square foot equivalent of one inch of water head. This reduces to

$$P = \frac{0.1573\ Qh}{e} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

which equation evidently applies to all fans. For the specific fan, $Q = 1$ $h = 1$, then

$$P_s = \frac{0.1573}{e}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

and, from (16) and (17),

$$P = P_s\ Qh \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

SPECIFIC TIP SPEED.

The tip speed of the specific fan, V_s, = feet per minute linear speed of the tip of the blades $= \frac{\pi D_s N_s}{12}$. Substituting for D_s and N_s their values from equations (12) and (15),

$$V_s = \frac{\pi D h^{\frac{1}{4}} N Q^{\frac{1}{2}}}{12 Q^{\frac{1}{2}} h^{\frac{3}{4}}} = \frac{\pi DN}{12 h^{\frac{1}{4}}}$$

but

$$\frac{\pi DN}{12} = V, \text{ then } V_s = \frac{V}{h^{\frac{1}{4}}} \quad . \quad . \quad . \quad . \quad (19)$$

and

$$V = V_s h^{\frac{1}{4}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

Equations (19) and (20) apply for a given fan or geometrically similar fans at constant efficiency.

COMPARISON OF FAN CHARACTERISTICS—UNIT FAN.

A comparison of the usual characteristics of quantity *vs.* head, quantity *vs.* efficiency, and quantity *vs.* power, is desirable in any study of the comparative performance of fans. This comparison

is best made by reducing the performance of the fans to a common basis of unit-fan size and speed. For this purpose the "unit fan" of a given type will be defined as a fan geometrically similar to a given fan of the type, but having a diameter of one foot and running at a speed of one thousand revolutions per minute. The characteristics of this unit fan may be determined from the known characteristics of the given fan or type in the following manner:

Notation for unit fan formulae.

d = diameter of given fan, feet.

d_u = diameter of unit fan, feet = 1.

n = speed of given fan, thousands or rev. per minute.

n_u = speed of unit fan, thousands or rev. per minute = 1.

h = head of given fan, inches of water.

h_u = head of unit fan, inches of water.

Q = quantity discharged, given fan, thousands of cubic feet per minute.

Q_u = quantity discharged, unit fan, thousands of cubic feet per minute.

P_u = horsepower of unit fan.

From equation (7) $Q = cnd^3$;

also $$Q_u = cn_u{d_u}^3;$$

then $$\frac{Q_u}{Q} = \frac{cn_u{d_u}^3}{cnd^3}, \text{ or}$$

$$Q_u = \frac{Q}{nd^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

From equation (8) $h = cn^2d^2$;

also $$h_u = cn_u{}^2d_u{}^2;$$

then $$\frac{h_u}{h} = \frac{cn_u{}^2d_u{}^2}{cn^2d^2} \text{ or}$$

$$h_u = \frac{h}{n^2d^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Equations (21) and (22) apply at constant efficiency. P_u may be obtained directly from equation (16), which applies to all fans.

$$P = \frac{0.1573\ Q h}{e}$$

then

$$P_u = \frac{0.1573\ Q_u\ h_u}{e} \quad . \quad . \quad . \quad . \quad . \quad (23).$$

At this point it will be well to distinguish between the two methods which have been derived for reducing the performance of fans to a comparable basis. There are four principal factors involved in performance, viz: diameter, speed, quantity and head. The specific-speed method reduces the performance to a basis of unit quantity and head and variable diameter and speed. The unit-fan method reduces the performance to a basis of unit diameter and speed and variable quantity and head. The unit-fan method may best be used to compare the operating characteristics of fans. The specific-speed method and formulae are best adapted to solving most of the numerical problems concerning the performance and selection of fans.

COMPUTATIONS FOR SPECIFIC AND UNIT FANS.

From the foregoing it may be gathered that the basis for making the computations required to cover the performance and design of a type of fan consists of data from a test of a certain fan of that type. The data should include quantity, head, speed and efficiency, and the diameter of the fan must be known. Computations have been made for the "specific fan" and the "unit fan" of a number of types of fans, test data for which have been kindly furnished by manufacturers and obtained from published results of tests.

Briefly, the method of performing the computations for a type of fan consists in selecting simultaneous values of quantity, head and efficiency from the characteristic curves obtained from a test of any fan of the type, and computing the values of N_s, D_s, V_s, h_u, Q_u, and P_u by means of the formulae previously derived. As an illustration, complete computations are given in Table I for fan type "A." The basis for these computations consist of the characteristic curves shown in Fig. 1

which were obtained from a test of a 120-inch diameter fan of type "A" running at 145 revolutions per minute. The significance and source of the items in Table I are:

Item 1 = Static efficiency, per cent., from curve Fig. 1 corresponding to Q.

Item 2 = Quantity discharged, thousands of cubic feet per minute, selected from curve Fig. 1.

Item 3 = Static head, inches of water, from curve Fig. 1, corresponding to Q.

Item 4 = $\overline{\text{Item 2}}^{\frac{1}{2}}$.

Item 5 = $\overline{\text{Item 3}}^{\frac{1}{2}}$.

Item 6 = $\overline{\text{Item 5}}^{\frac{1}{3}}$.

Item 7 = $\overline{\text{Item 6}}^{3}$.

Item 8 = Specific speed, $N_s = \frac{NQ^{\frac{1}{2}}}{h^{\frac{3}{4}}} = \frac{145 \times \text{Item 4}}{\text{Item 7}}$.

Item 9 = Specific diameter; $D_s = \frac{Dh^{\frac{1}{4}}}{Q^{\frac{1}{2}}} = \frac{120 \times \text{Item 6}}{\text{Item 4}}$.

Item 10 = Specific tip speed; $V_s = \frac{V}{h^{\frac{1}{2}}} = \frac{\pi \times 145 \times 120}{12 \times \text{Item 5}}$

$= \frac{4{,}606}{\text{Item 5}}$.

Item 11 = Unit quantity, $Q_u = \frac{Q}{nd^3}$, in which

Q_u = thousands of cubic feet per minute discharged by unit fan,

n = thousands of rev. per minute = 0.145,

d = diameter, feet, = 120 ÷ 12 = 10 feet,

$$Q_u = \frac{Q}{0.145 \times 1{,}000} = \frac{\text{Item 2}}{145}.$$

Item 12 = Unit head, $h_u = \frac{h}{n^2d^2}$, in which

n = thousands of rev. per minute = 0.145,

d = diameter, feet = 10,

$$h_u = \frac{h}{0.145 \times 0.145 \times 100} = \frac{\text{Item 3}}{2.1}.$$

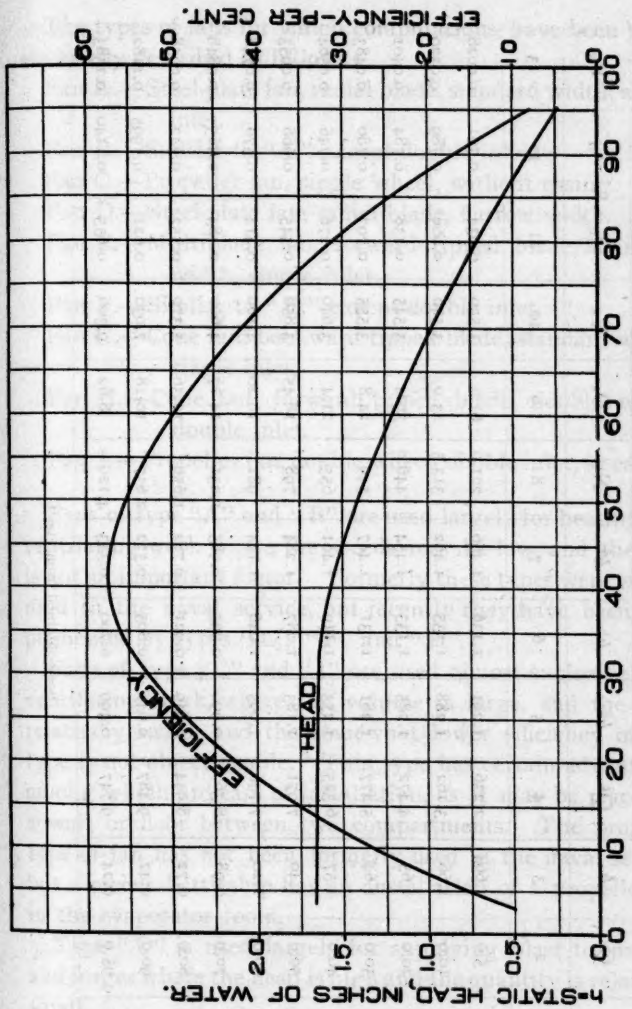


FIG. 1.—CONSTANT SPEED CHARACTERISTICS OBTAINED FROM TEST OF 120-INCH STEEL-PLATE FAN, TYPE A, AT 145 R.P.M. BASIS FOR COMPUTATIONS OF TABLE I.

$$\text{Item 13} = \text{Unit power, } P_u = \frac{Q_u\, h_u \times 0.1573}{e}$$

$$= \frac{\text{Item 11} \times \text{Item 12} \times 0.1573}{\text{Item 1} \times \frac{1}{100}}.$$

TABLE I.—COMPUTATION OF SPECIFIC FAN AND UNIT FAN. DATA FOR FAN, TYPE "A."

	e	Q	h	$Q^{\frac{1}{2}}$	$h^{\frac{1}{2}}$	$h^{\frac{1}{3}}$	$h^{\frac{2}{3}}$	N_s	D_s	V_s	Q_u	h_u	P_u
Item.	1	2	3	4	5	6	7	8	9	10	11	12	13
1	15.5	5.0	1.664	2.236	1.291	1.136	1.466	221.0	61.0	3,530	0.034	0.791	0.0028
2	26.5	10.0	1.66	3.162	1.288	1.135	1.462	313.5	43.0	3,540	0.069	0.789	0.0320
3	42.5	20.0	1.65	4.472	1.285	1.133	1.454	446.0	30.4	3,545	0.138	0.784	0.0401
4	52.8	30.0	1.64	5.477	1.281	1.131	1.447	549	24.8	3,555	0.207	0.780	0.0483
5	57.3	40.0	1.57	6.325	1.253	1.119	1.401	655	21.4	3,640	0.276	0.746	0.0564
6	55.3	50.0	1.40	7.071	1.183	1.087	1.284	798	18.45	3,850	0.345	0.666	0.0653
7	49.6	60.0	1.20	7.746	1.095	1.046	1.144	981	16.20	4,160	0.414	0.570	0.0748
8	41.7	70.0	0.98	8.367	0.990	0.995	0.985	1,231	14.27	4,600	0.483	0.465	0.0848
9	31.0	80.0	0.72	8.944	0.849	0.921	0.782	1,659	12.36	5,370	0.552	0.342	0.0958
10	16.5	90.0	0.42	9.487	0.648	0.805	0.522	2,637	10.18	7,030	0.621	0.1995	0.1182
11	8.5	95.0	0.24	9.747	0.490	0.700	0.343	4,120	8.62	9,300	0.655	0.1140	0.1383

Basis obtained from Fig. 1.

The types of fans for which computations have been made are briefly described as follows:

Fan A.—Steel-plate fan, radial blade, standard width, single inlet.

Fan B.—Similar to "A," except double inlet.

Fan C.—Propeller fan, single wheel, without casing.

Fan D.—Steel-plate fan, radial blade, narrow width.

Fan E.—Multiblade fan, forward tipped blade, standard width, single inlet.

Fan F.—Similar to "E," except double inlet.

Fan G.—Cone fan, backward tipped blade, standard width, single inlet.

Fan H.—Cone fan, forward tipped blade, double width, double inlet.

Fan I.—Propeller fan, double wheel, double inlet, in casing.

Fans of type "A" and "B" are used largely for heating and ventilating work where the speeds may be low, and the size is not an important factor. Formerly these types were widely used in the naval service, but recently they have been supplemented by types "E," "F" and "H".

Fans of type "C" and "I" are used almost exclusively for ventilating work where the volume is large, and the head relatively small, and the somewhat lower efficiency of this type is not objectionable. This type has certain advantages, among which are ease of installation, as it may be placed in a wall or floor between two compartments. The propeller type of fan has not been formerly used in the naval service, but a recent battleship has an installation of a propeller fan in the evaporator room.

Type "D" is used largely for supplying blast to furnaces and forges where the head is high and the quantity is relatively small.

Types "E" and "F" are used for heating and ventilating, having the advantages over the plate fan for this service of a smaller size, flat quantity-head characteristic, and slightly better efficiencies. The superiority of the multiblade fan over

Basis obtained from Fig. 1.

the steel-plate fan for all installations is frequently overestimited, however. It will be shown in this paper how the particular requirements of each installation must be considered in selecting the most suitable type of fan.

Fans of type "G" are used for forced-draft work, and in the naval service are used largely to supply forced draft on destroyers. Fans of type "H" are used for ventilating and forced-draft duty, and in the naval service are used frequently for forced draft in battleships.

The results of the computations for these nine types of fans have been presented in the form of curves on Figs. 2 to 9, inclusive. Each of these figures contain a group of similar curves for all of the types considered. Before illustrating the use of these curves and the previously derived formulae in solving problems, it will be well to make some general observations regarding the curves.

Curves on Figure 2 show the relation between specific speed and efficiency for the several types of fans. From formula (11) it is seen that, for any definite values of quantity and head, the actual speed of a fan is proportioned to its specific speed. It is also an evident fact that any type of fan may be used to produce any desired values of quantity and head, within certain limits, if the fan be made of the proper size and run at the proper speed. From Fig. 2 it is seen, however, that each type of fan operates at its best efficiency at only one value of specific speed, and therefore at only one particular actual speed, for certain desired values of quantity and head. At speeds either greater or less than this best speed the efficiency falls off.

This shows immediately the importance of selecting the proper speed at which a fan of a given type is to operate in order to produce a given quantity and head at good efficiency. It is further seen that the various types of fans reach their best efficiencies at widely different values of specific speed, and therefore of actual speed. These properties of the specific speed curves suggest their use in selecting fan types and in solving problems involving speed. The specific speed curves

also provide a basis for the classification of fans according to the speed at which the maximum efficiency occurs.

Thus, fan type "D" is a "low specific speed" fan because its maximum efficiency occurs at a specific speed of 640 revolutions per minute. Types "A" and "B" belong to a class whose specific speed varies from 600 to 800 at good efficiencies, and may be called medium speed types. Types "E" and "F" are most efficient at values of $N_s = 900$, and types "C" and "I" are extremely high specific speed fans. In general the low specific speed types are more efficient. Certain types of fans have efficiencies over a wide range which are nearly equal to their best efficiency, and other types have a rather limited range in speeds over which the efficiency is good.

At this point a careful distinction must be made between the actual speed of a fan and its specific speed. It is true that the actual speed is proportional to the specific speed, as before stated, but it is also observed that fans with low specific speeds may operate at high actual speeds. An inspection of the specific speed formula, $N = N_s \frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}}$, shows that for relatively high values of head it is necessary to use a type of fan with a low value of N_s in order to prevent the actual speed from reaching excessive limits. For this reason low specific speed fans are used for high heads, and high specific speed fans for low heads.

Figure 3 shows the relation between "specific diameter" and efficiency for the several types of fans. From formulae (14) it is seen that the actual diameter of a fan of a certain type is proportional to its specific diameter for a given quantity and head. This group of curves shows that each type of fan operates at its best efficiency only at one value of D_s and therefore at only one actual diameter or size for given values of quantity and head. For diameters larger or smaller than this the efficiency decreases. This shows the importance of designing for correct size, and suggests that the curves may be used to solve problems involving diameter or size, as well as to classify the types of fans according to size. Thus, "H"

and "I" may be classified as small fans, and "D" is a large diameter fan, for the same service.

From Fig. 2 and 3 it is noticed that there are two different specific speeds at any efficiency, except the maximum, and two corresponding values of specific diameter. The relation between specific speed and specific diameter is shown on Fig. 4, where N_s is plotted against D_s, independent of efficiency. This plot shows that each value of N_s has a corresponding value of D_s, the lower value of N_s corresponding to the higher value of D_s. That is, as the size of a fan is increased the speed is decreased, the requirements as to quantity and head remaining constant. Therefore in working with curves 2 and 3 when there is a choice between two diameters and two corresponding speeds, at a given efficiency, it is only necessary to bear in mind that the larger diameters must be associated with the lower speeds and vice versa.

It is observed that all the curves of Fig. 4 appear to be of the hyperbolic form. If these curves were rectangular hyperbolas their equation would be D_sN_s = constant, and this would correspond to constant tip speed of the specific fan. Lines of constant specific tip speed have been drawn in on Fig. 4 for the purpose of analyzing this property of the curves. Fig. 4 may be used to compare directly the relative sizes of various types of fans required for fixed conditions of quantity, head and speed. Likewise they may be used to show relative speeds required by various types for fixed conditions of quantity, head and diameter.

Tip speed may be made the basis for the classification of fans. Equation 20 shows that, at any given head, the actual tip speed of a fan is proportional to its specific tip speed, V_s. From Fig. 5 it is seen that fans from types E and F have the lowest tip speed, V_s, being about 3,000 feet per minute at the best efficiency. Types D and H have a specific tip speed of about 3,400 and types G and I have the highest tip speed, reaching between 6,000 and 7,000 at good efficiencies. For any desired head each type of fan must operate at only the

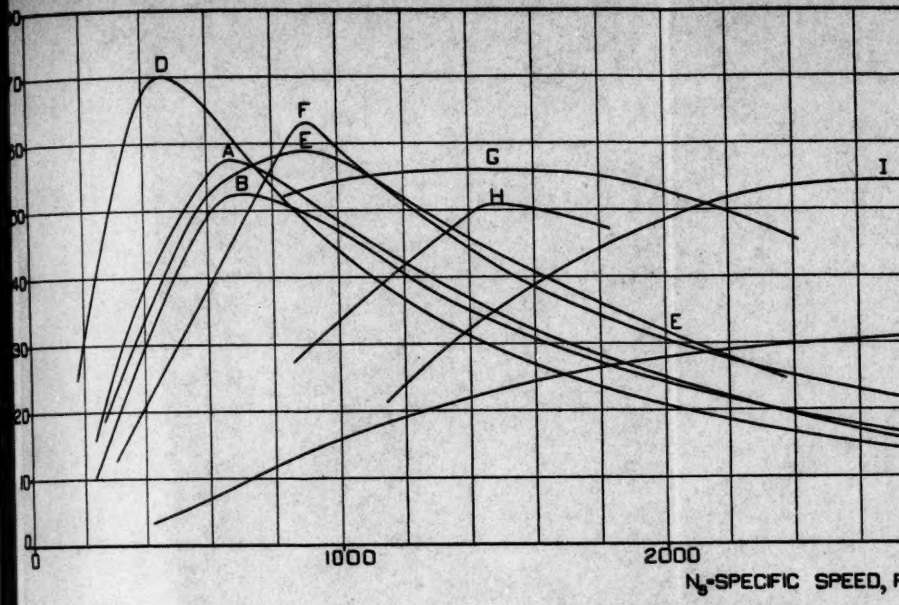


FIG. 2.—SPECIFIC SPEED.—EFF

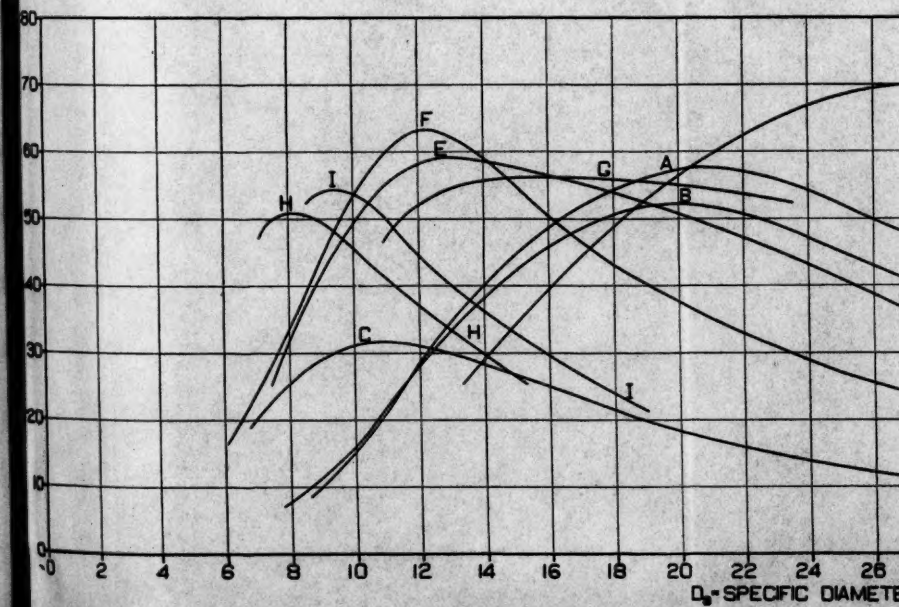
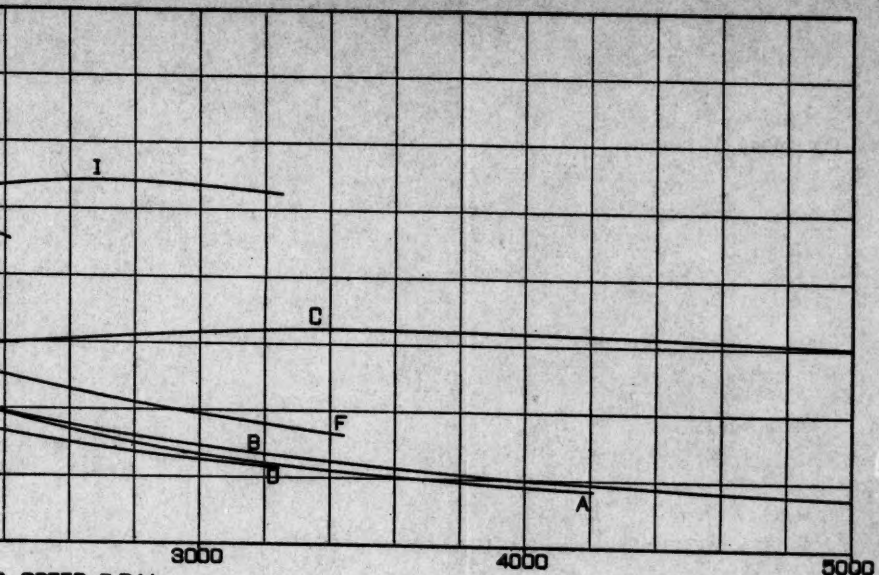


FIG. 3.—SPECIFIC DIAMETER.—E



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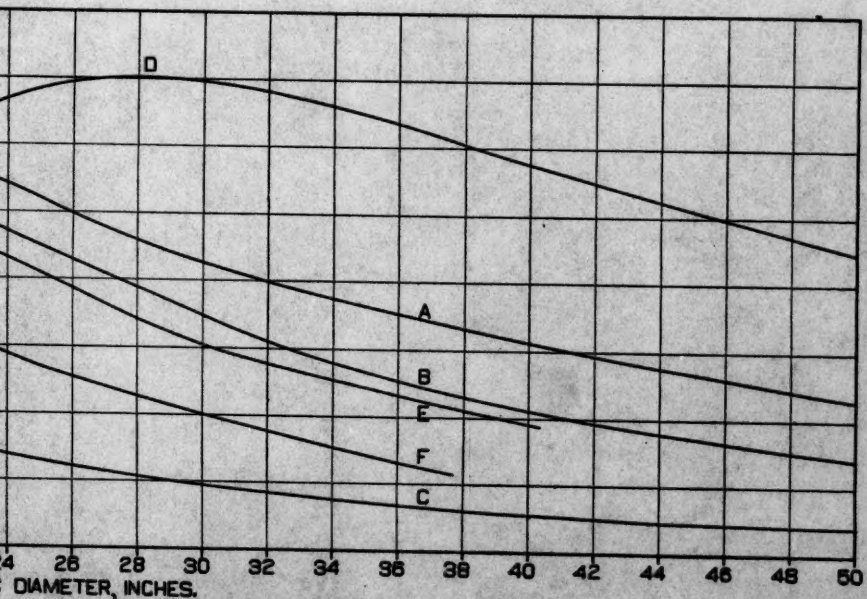
ED.—EFFICIENCY CURVES.



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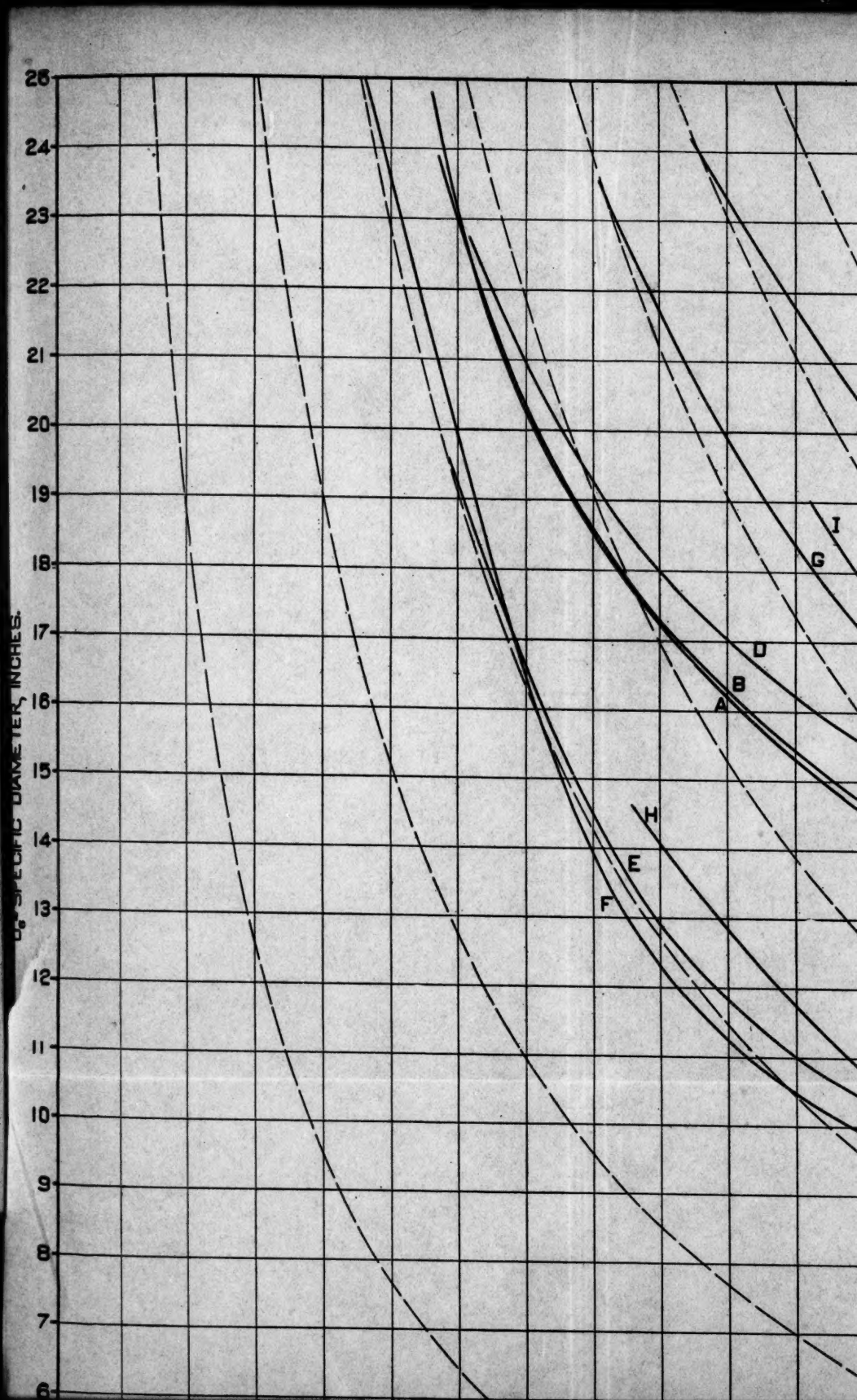
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D. SPECIFIC DIAMETER, INCHES.
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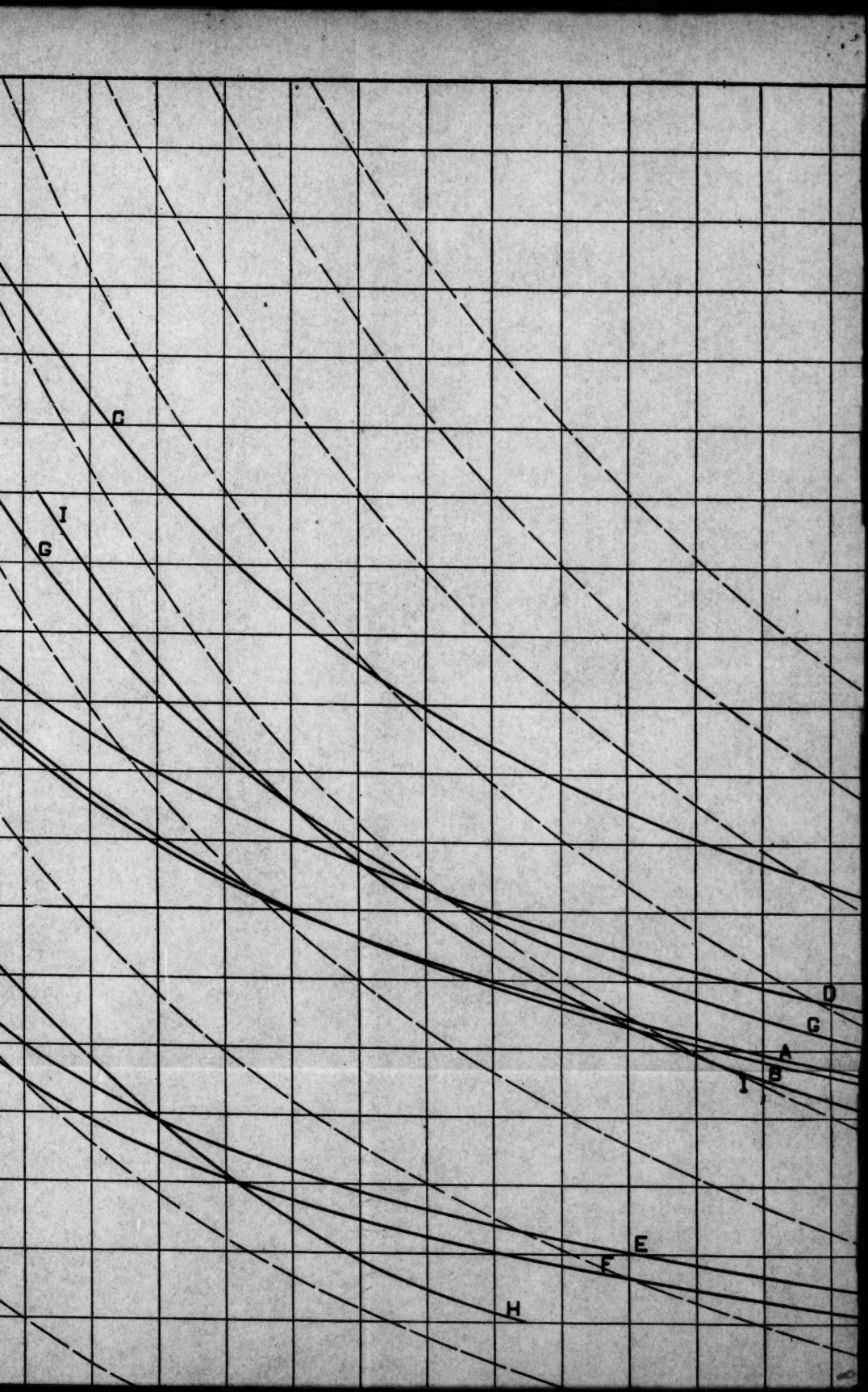
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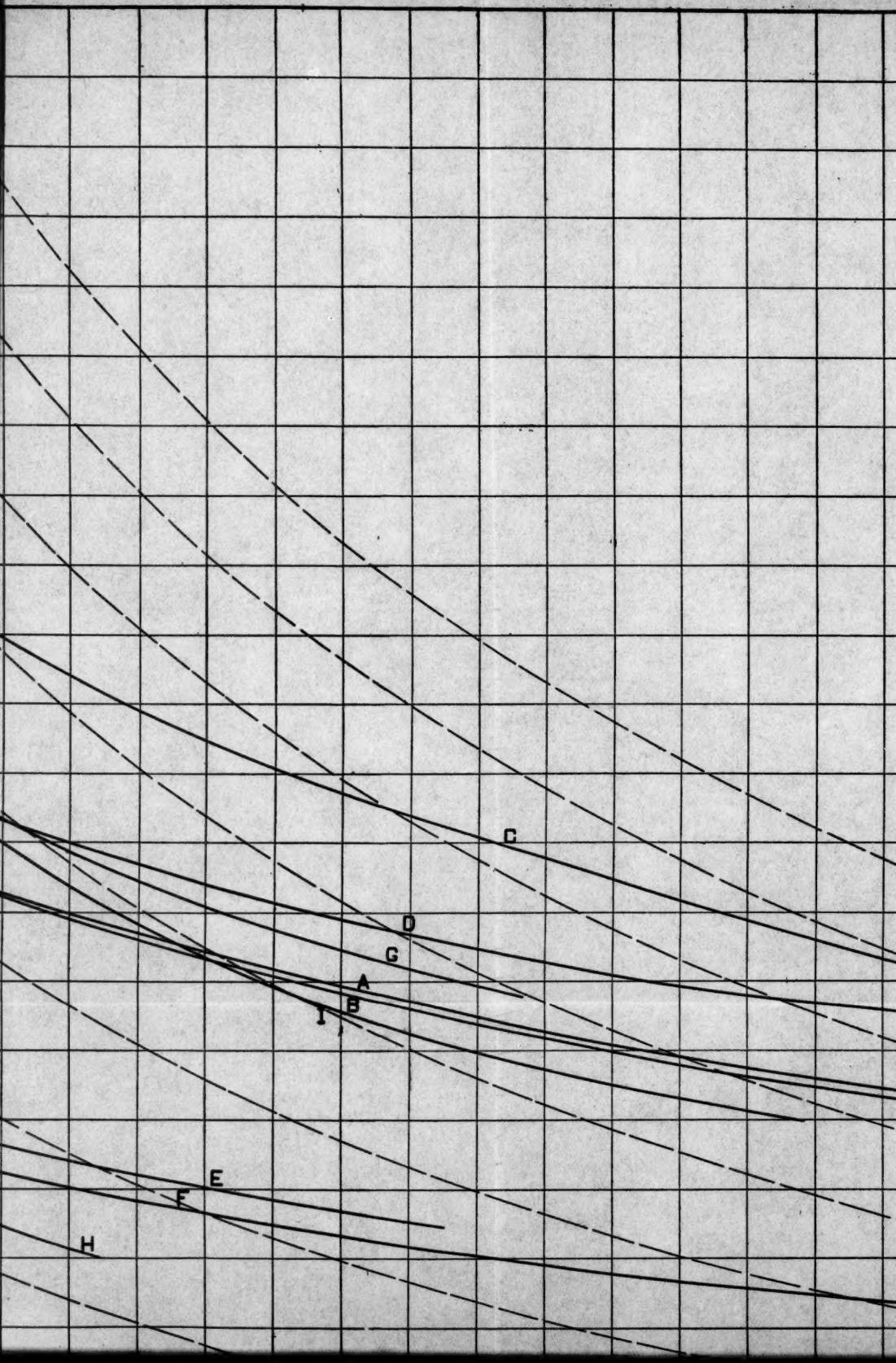
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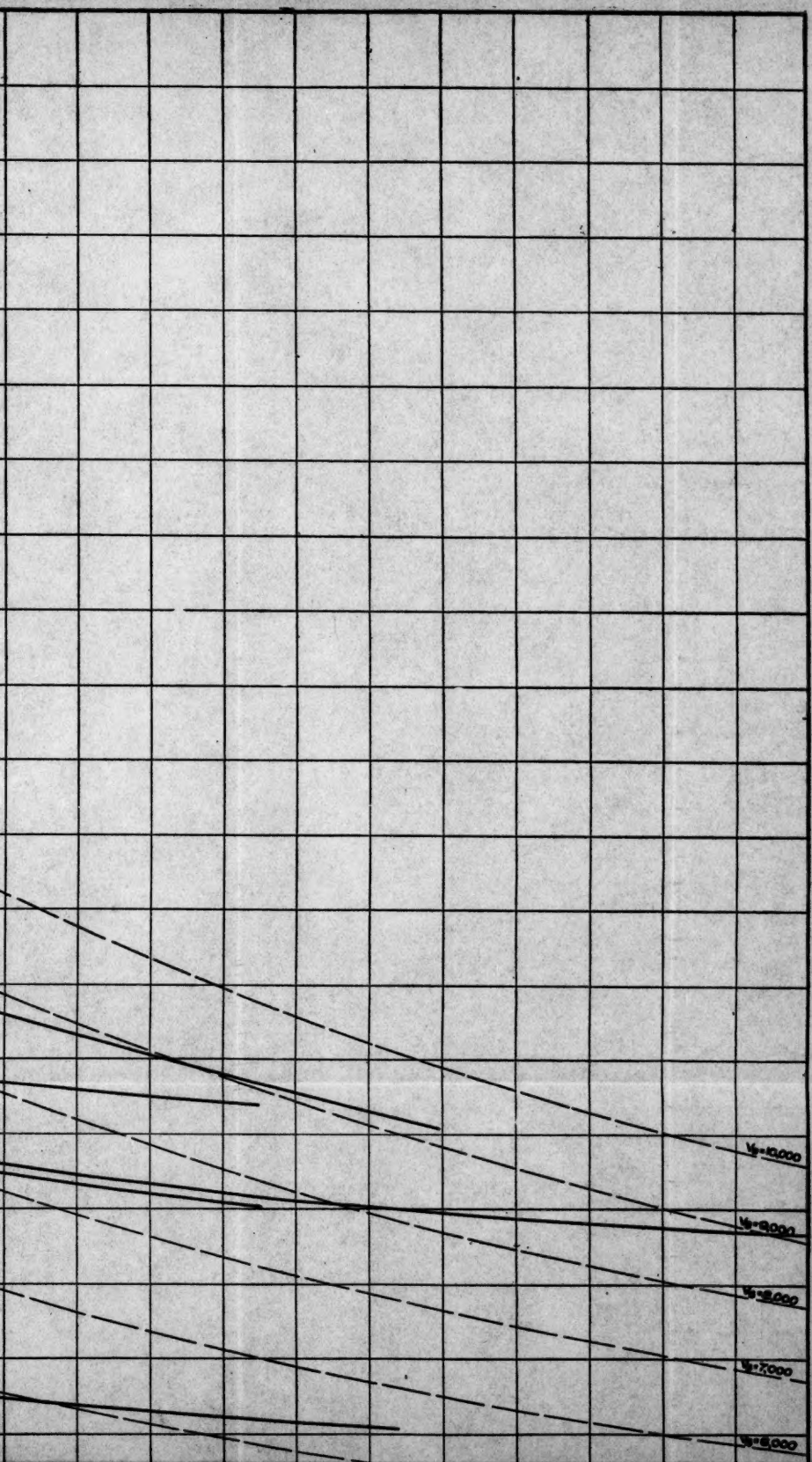
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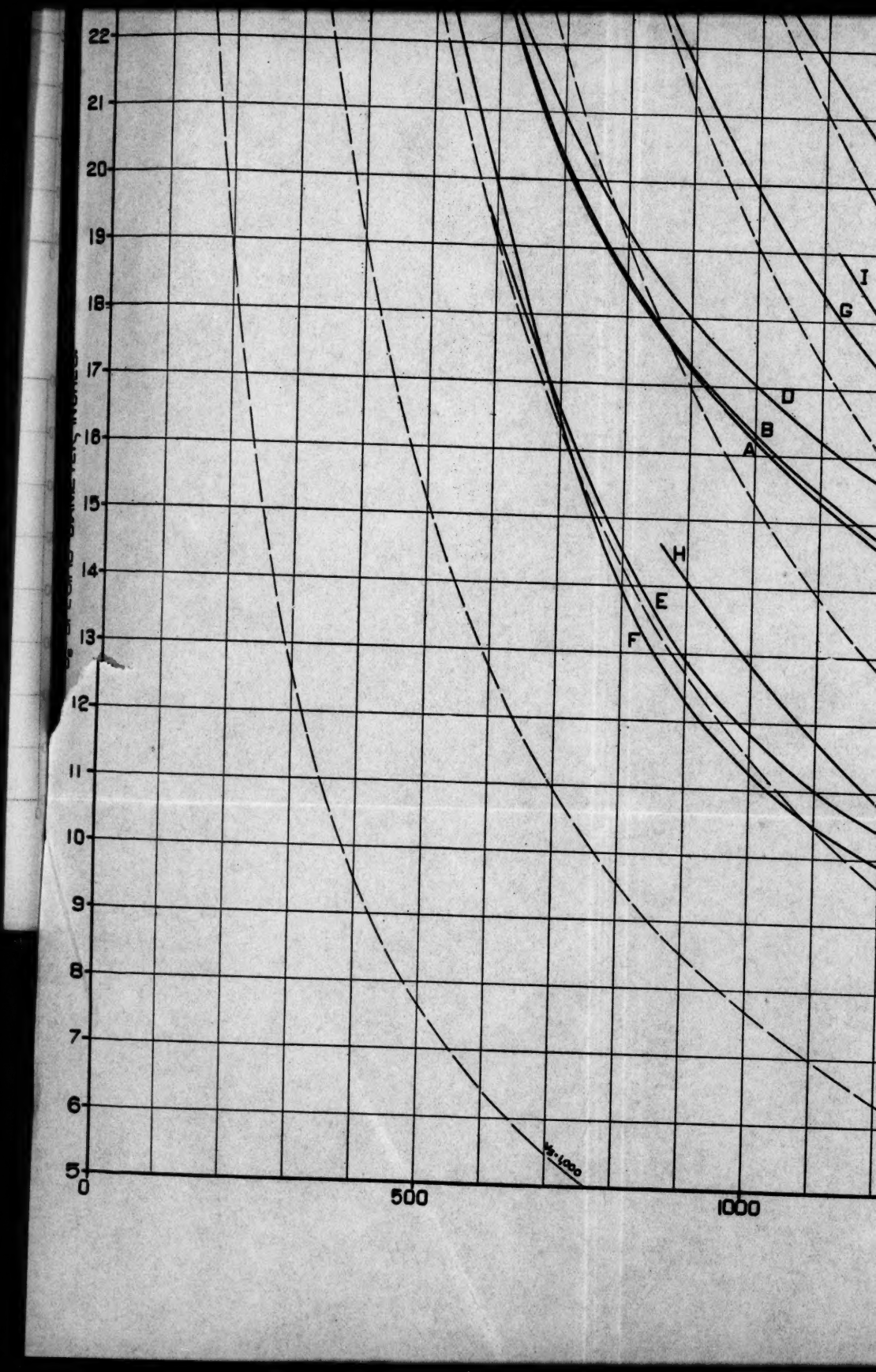
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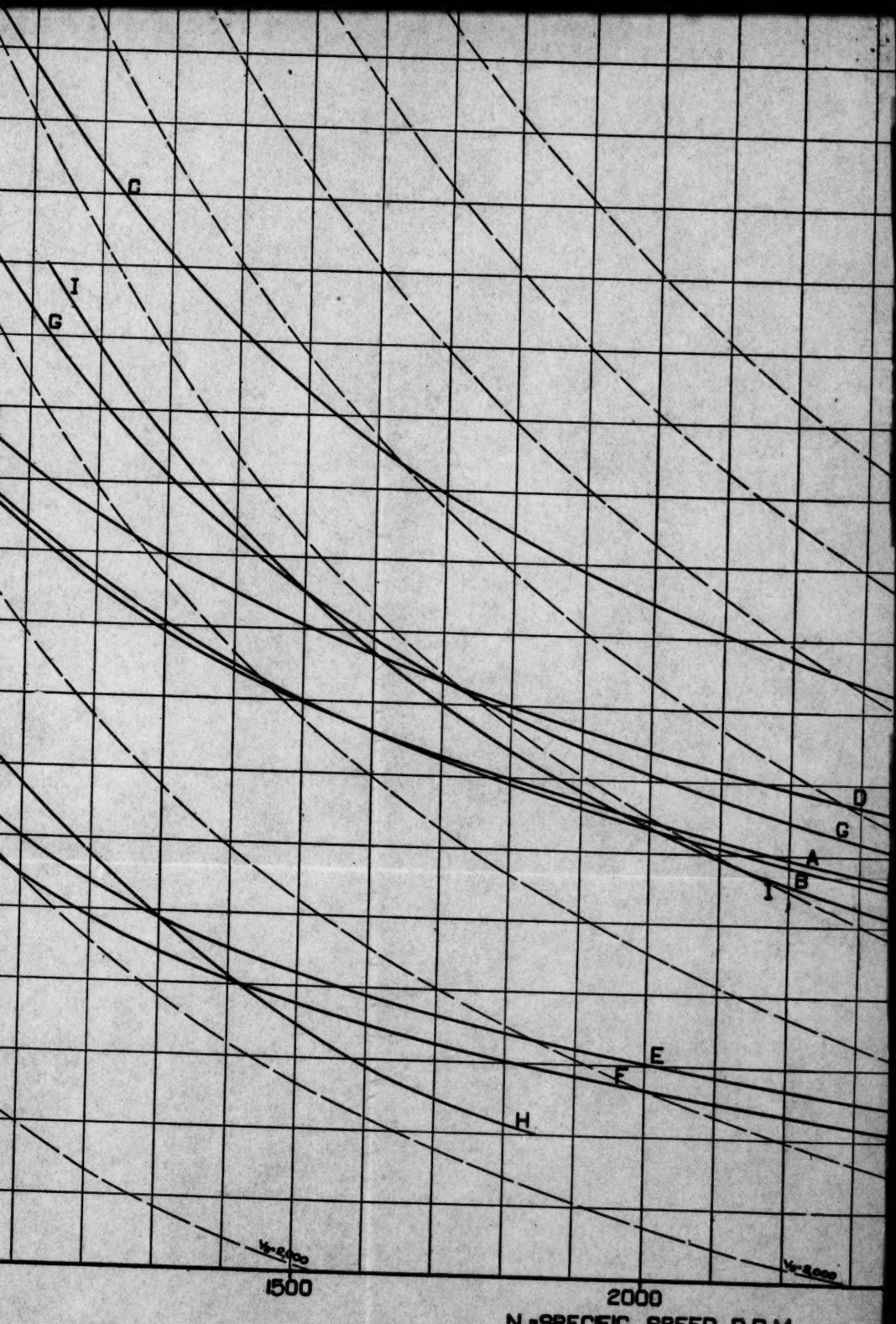


FIG. 4.—SPECIFIC SPEED.—SPECIFIC DIAMETER CU

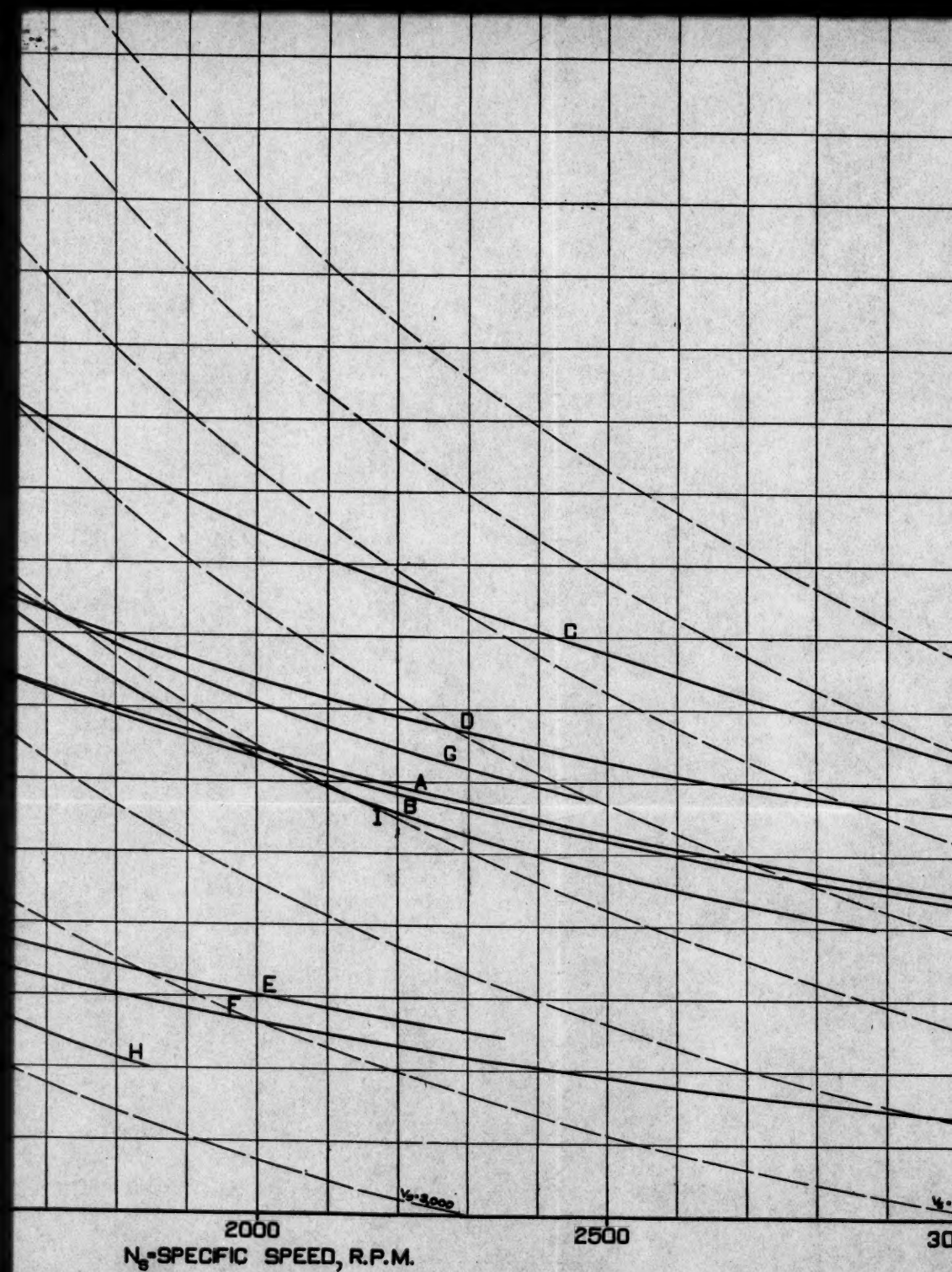
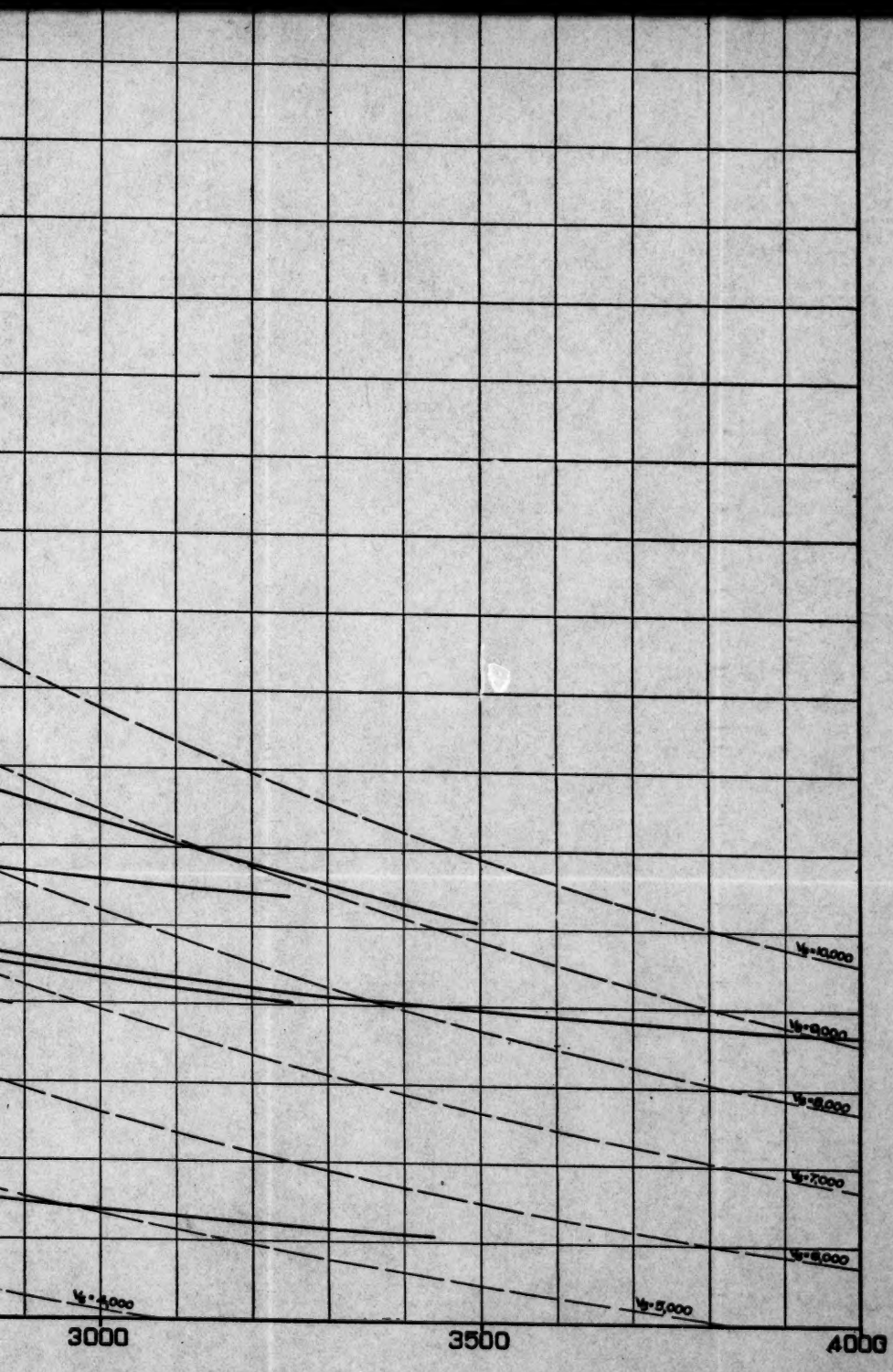


FIG. 4.—SPECIFIC SPEED.—SPECIFIC DIAMETER CURVES.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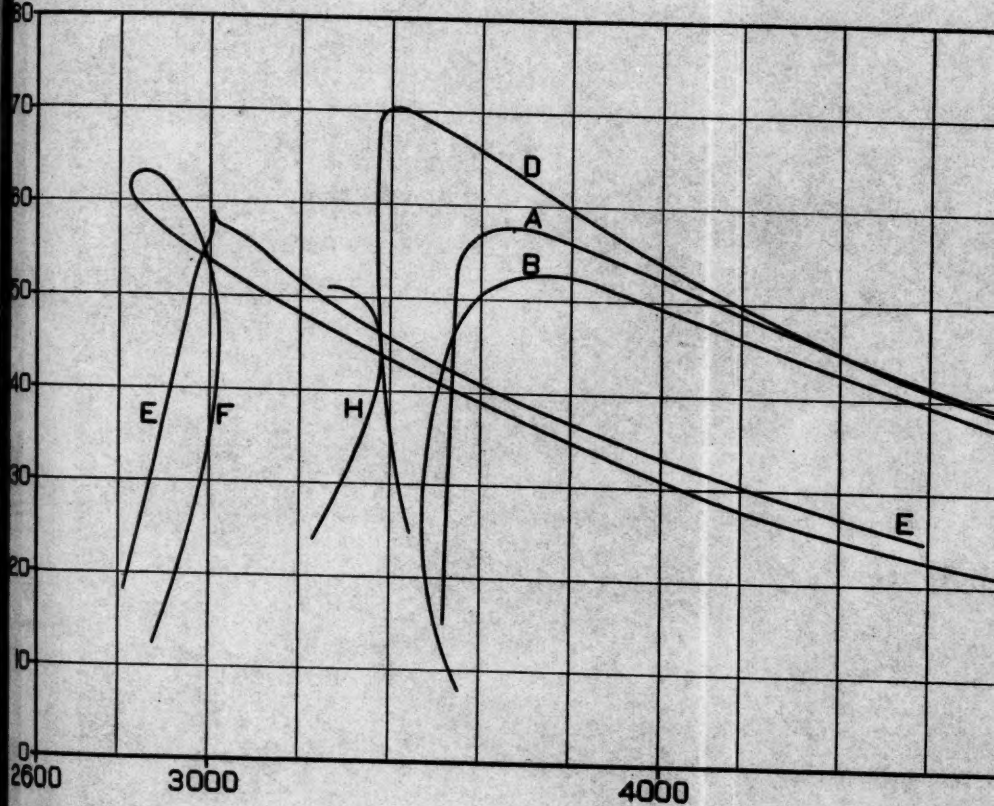
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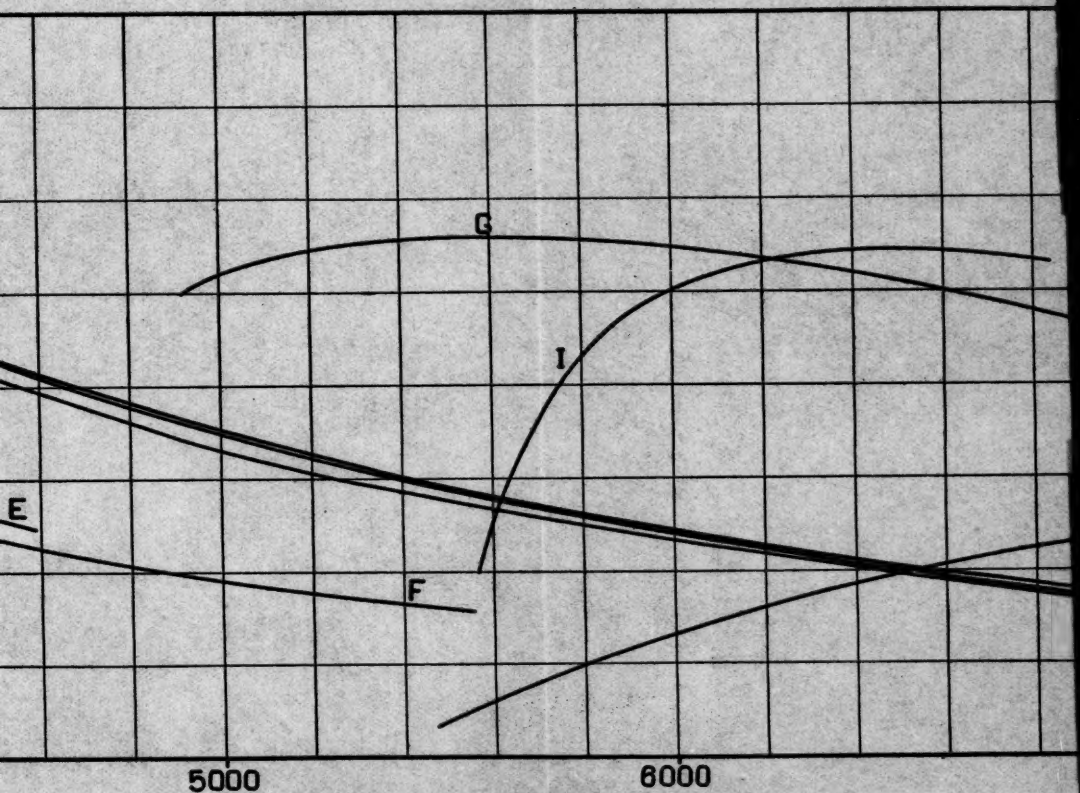


FIG. 5.—SPECIFIC TIP SPEED.—EFFICIENCY CURVES.

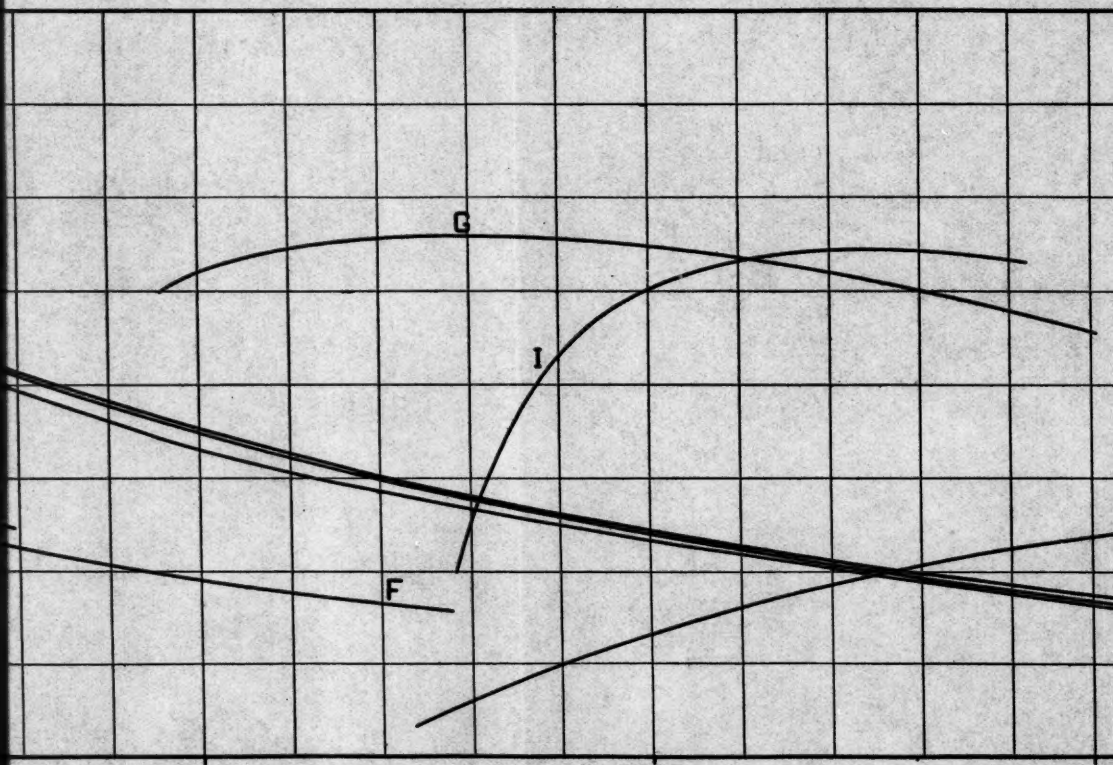
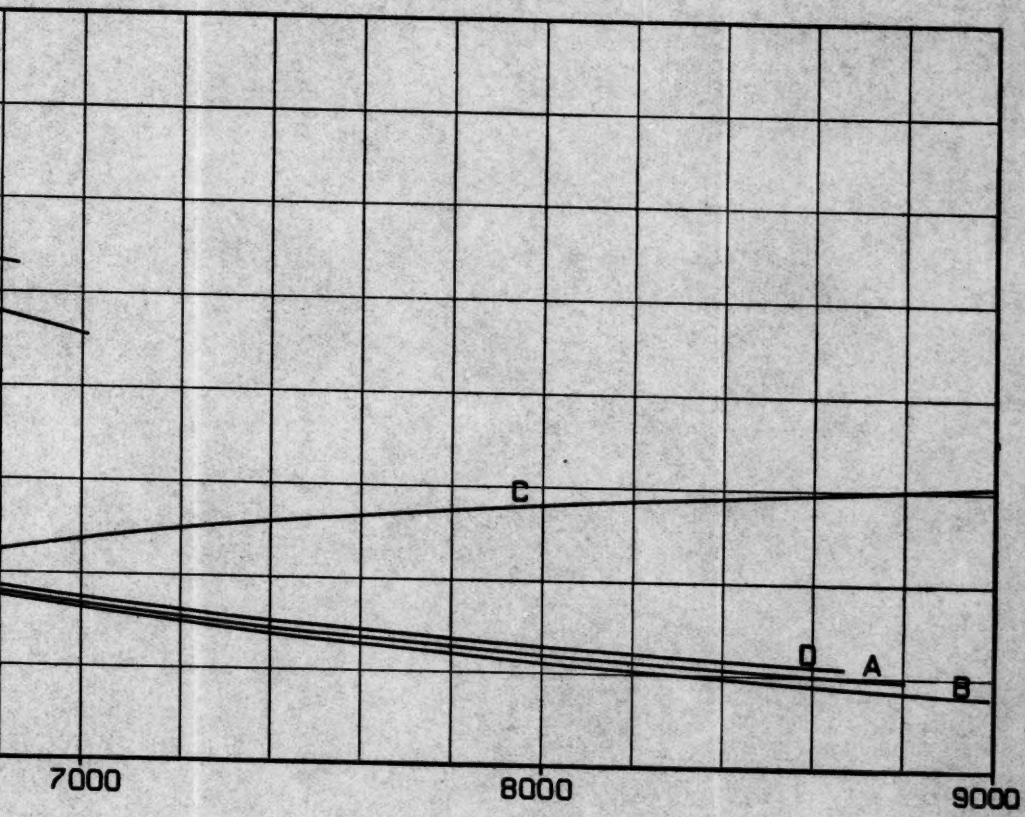


FIG. 5.—SPECIFIC TIP SPEED.—EFFICIENCY CURVES.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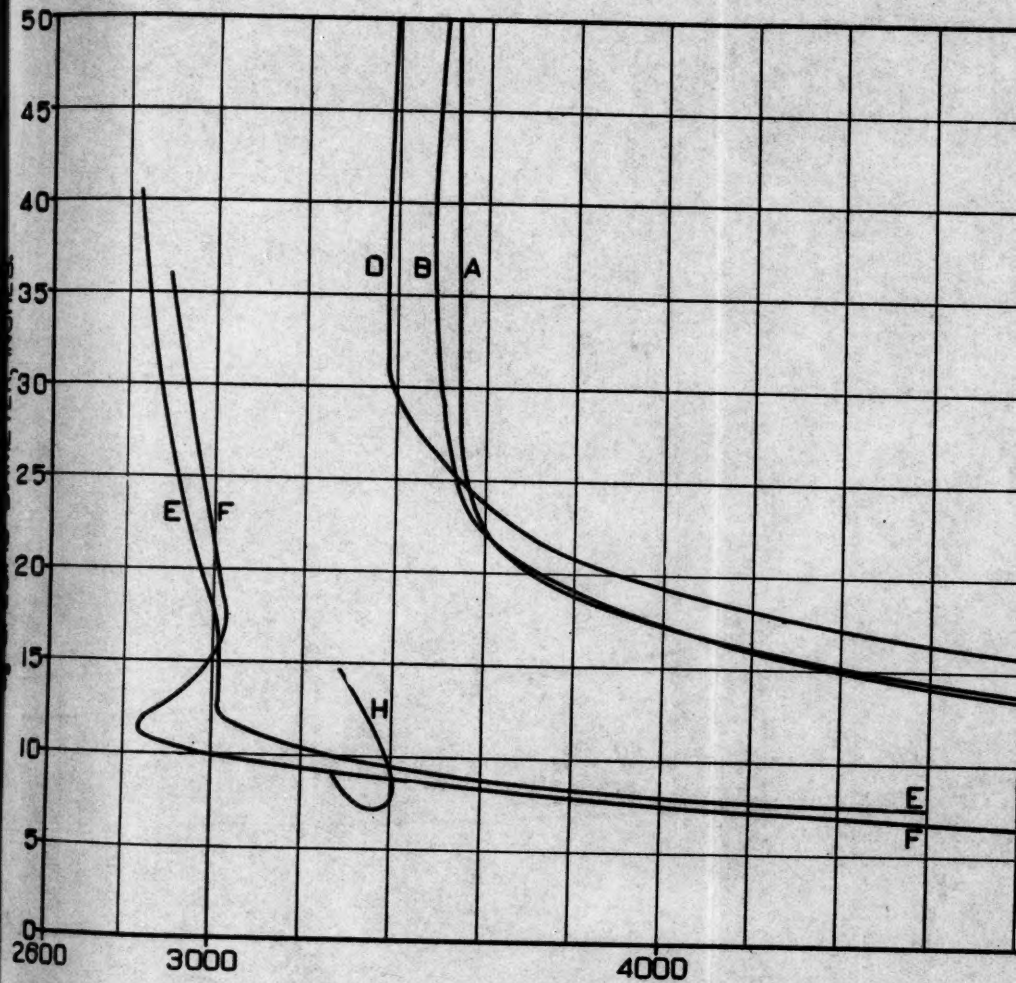
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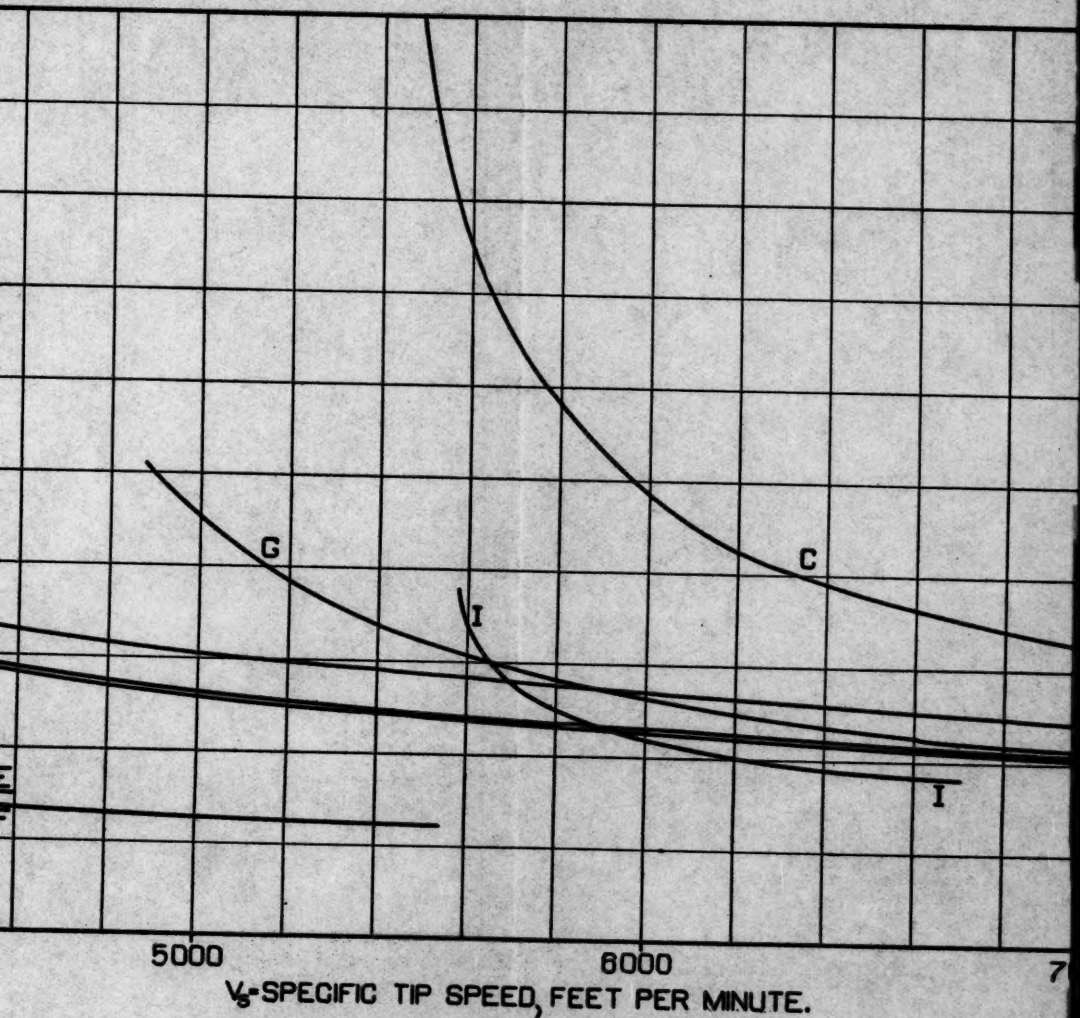


FIG. 6.—SPECIFIC TIP SPEED.—SPECIFIC DIAMETER CURVES.

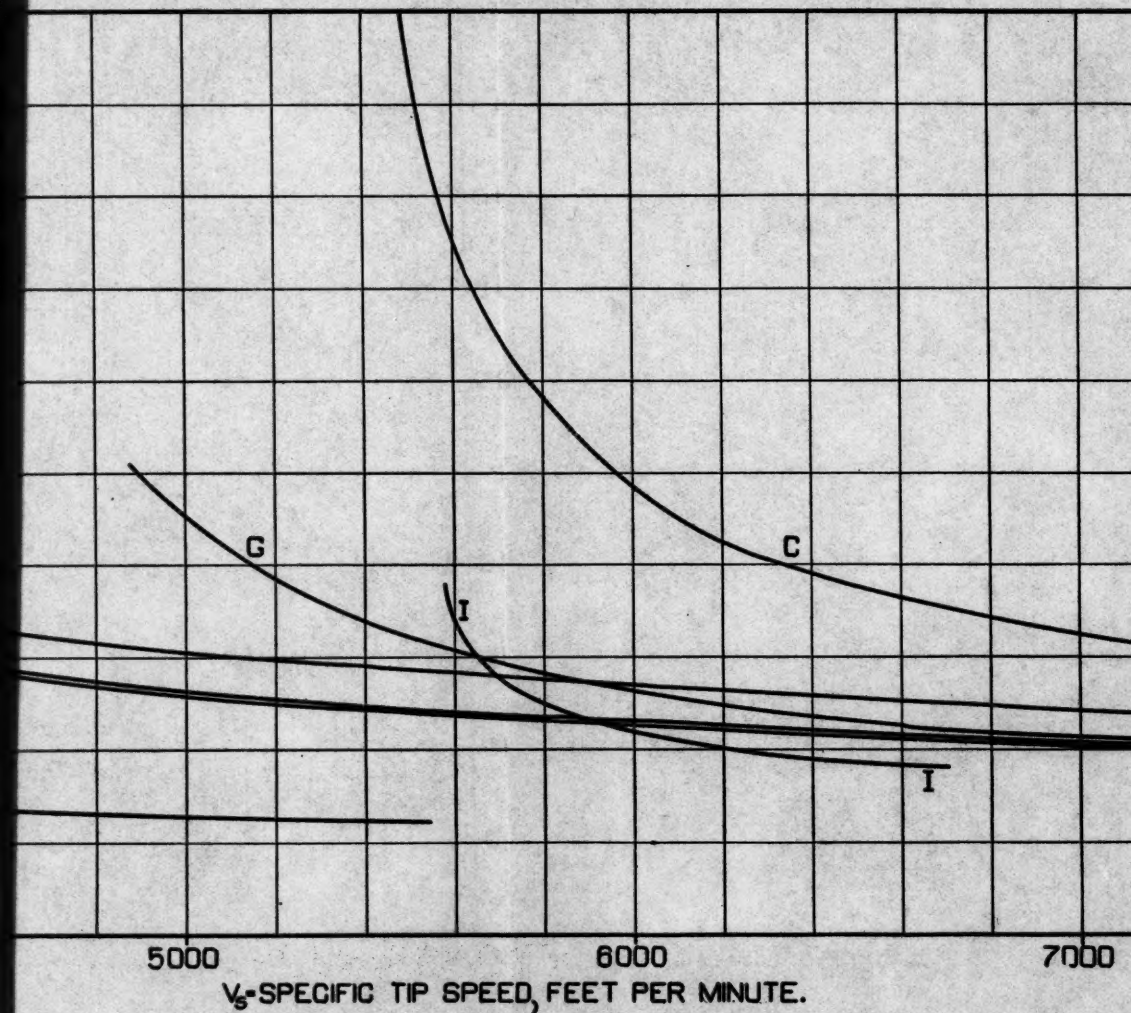
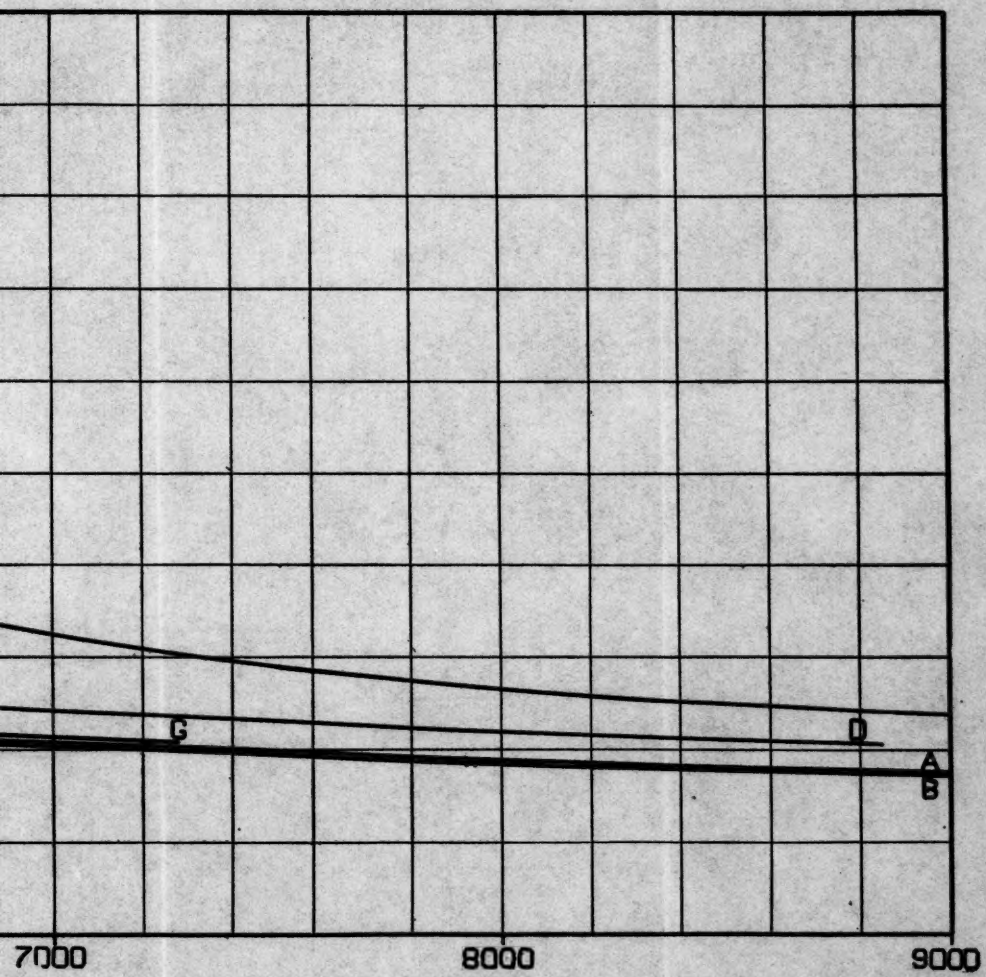


FIG. 6.—SPECIFIC TIP SPEED.—SPECIFIC DIAMETER CURVES.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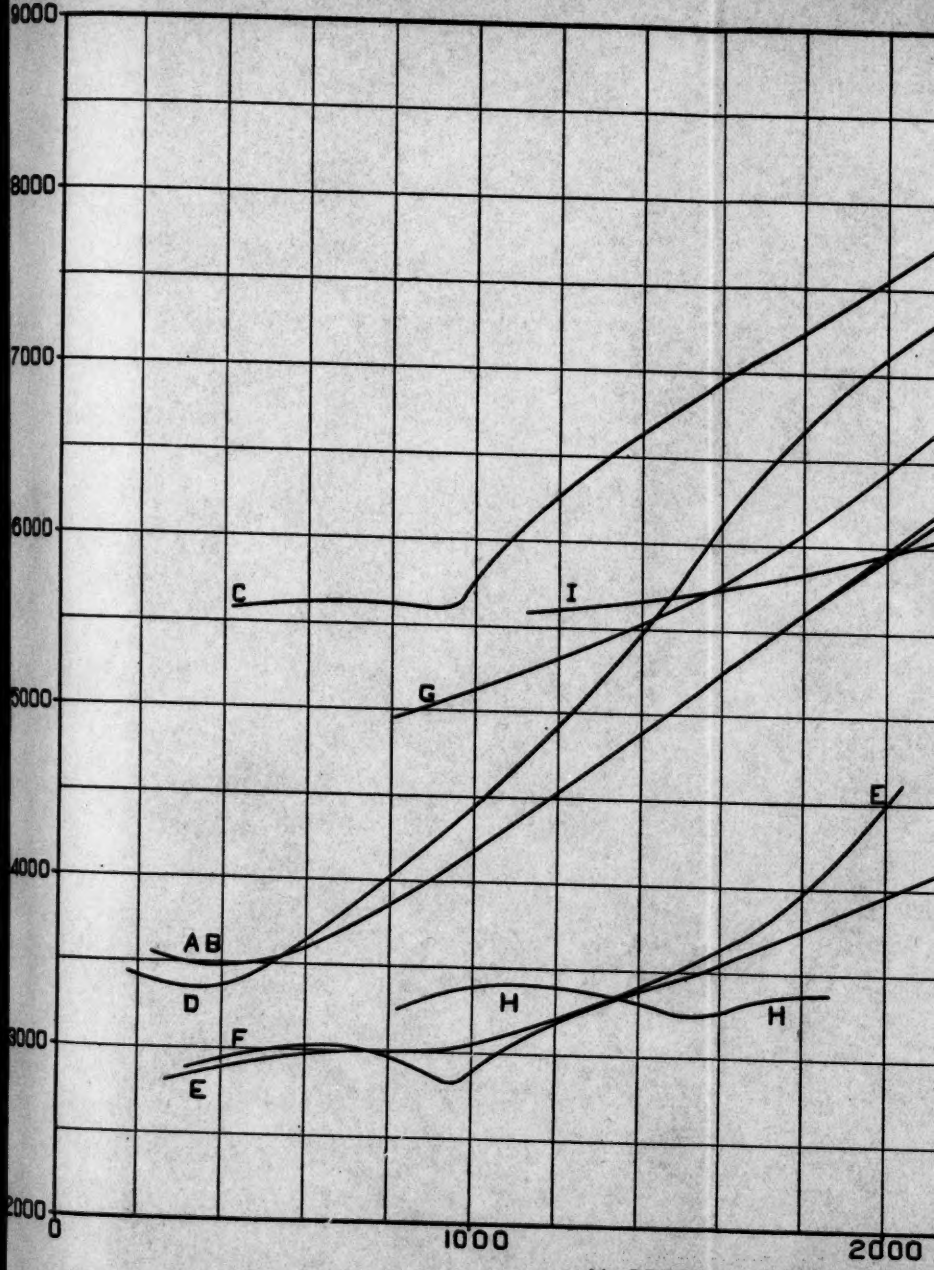
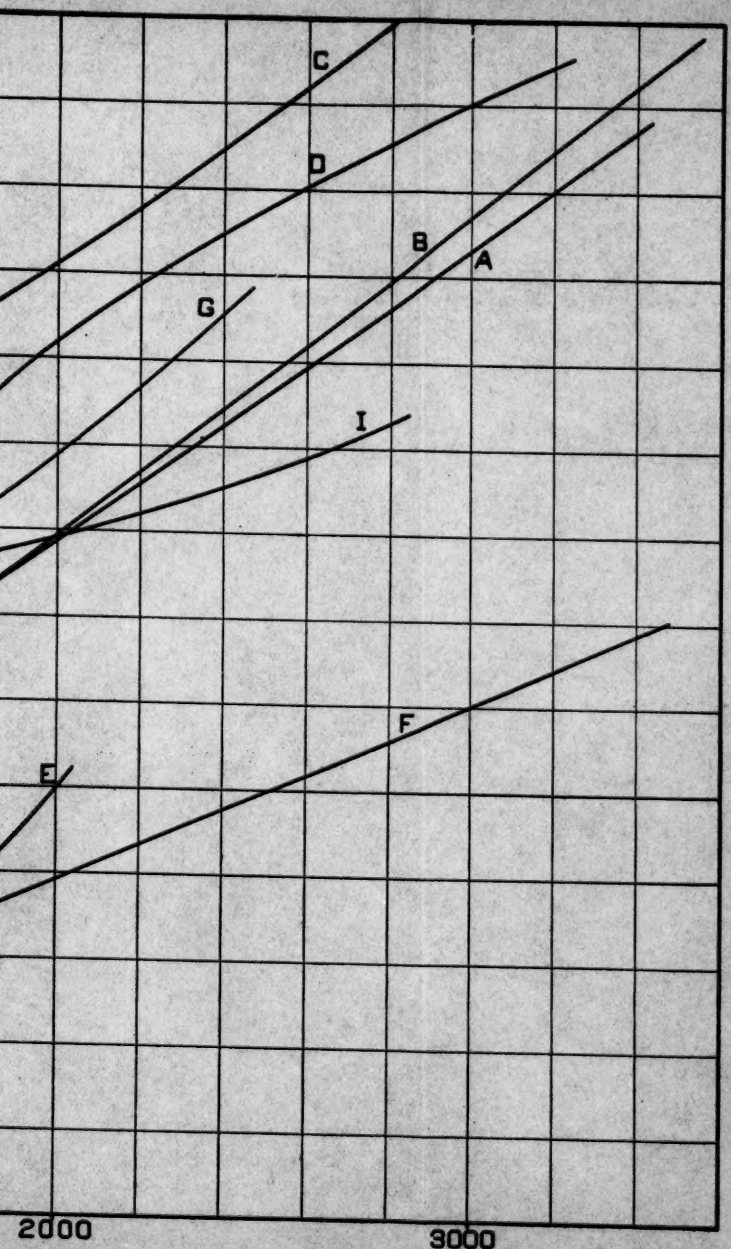


FIG. 7.—SPECIFIC SPEED.—SPECIFIC TIP-SPEE



TIP-SPEED CURVES.

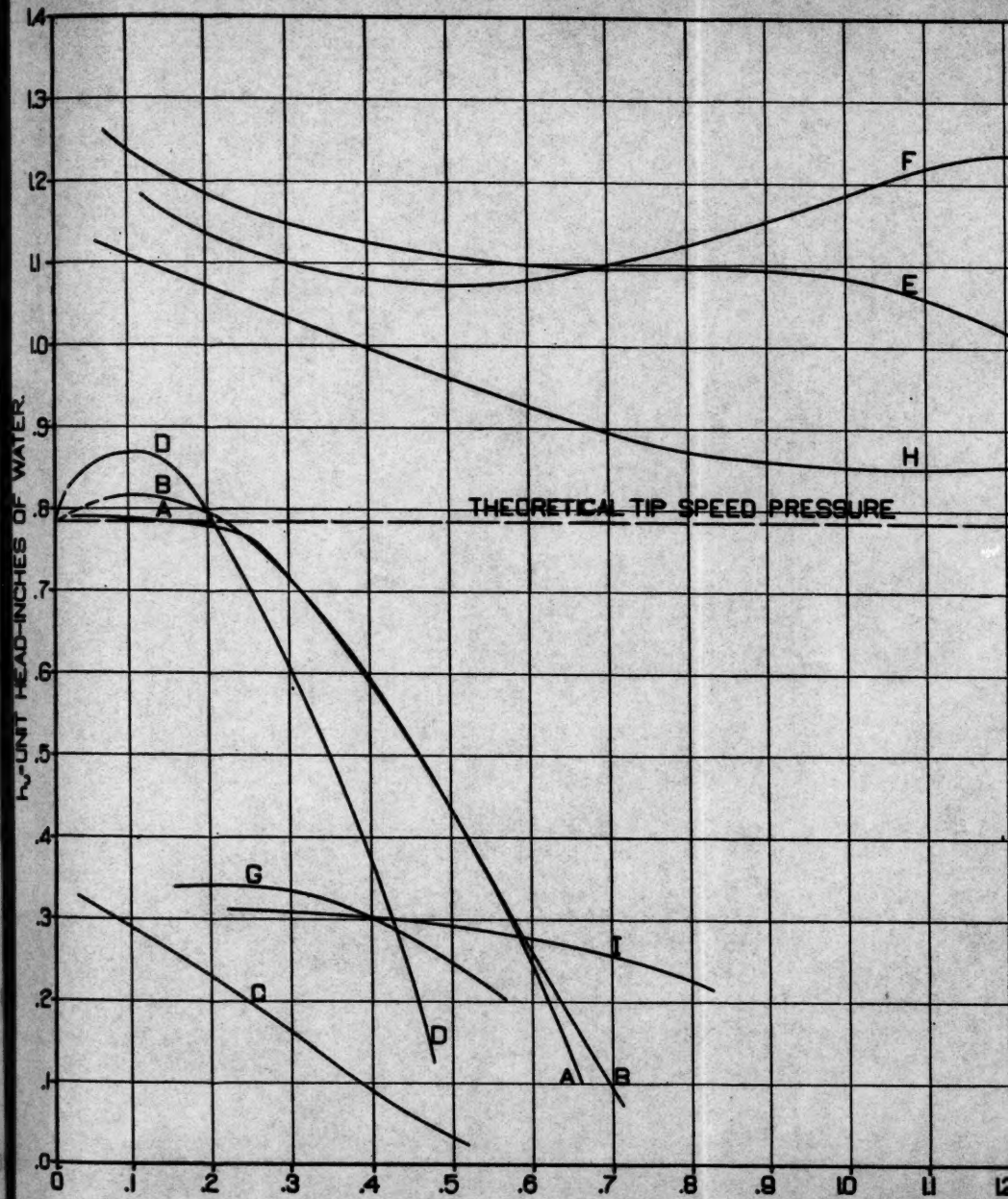
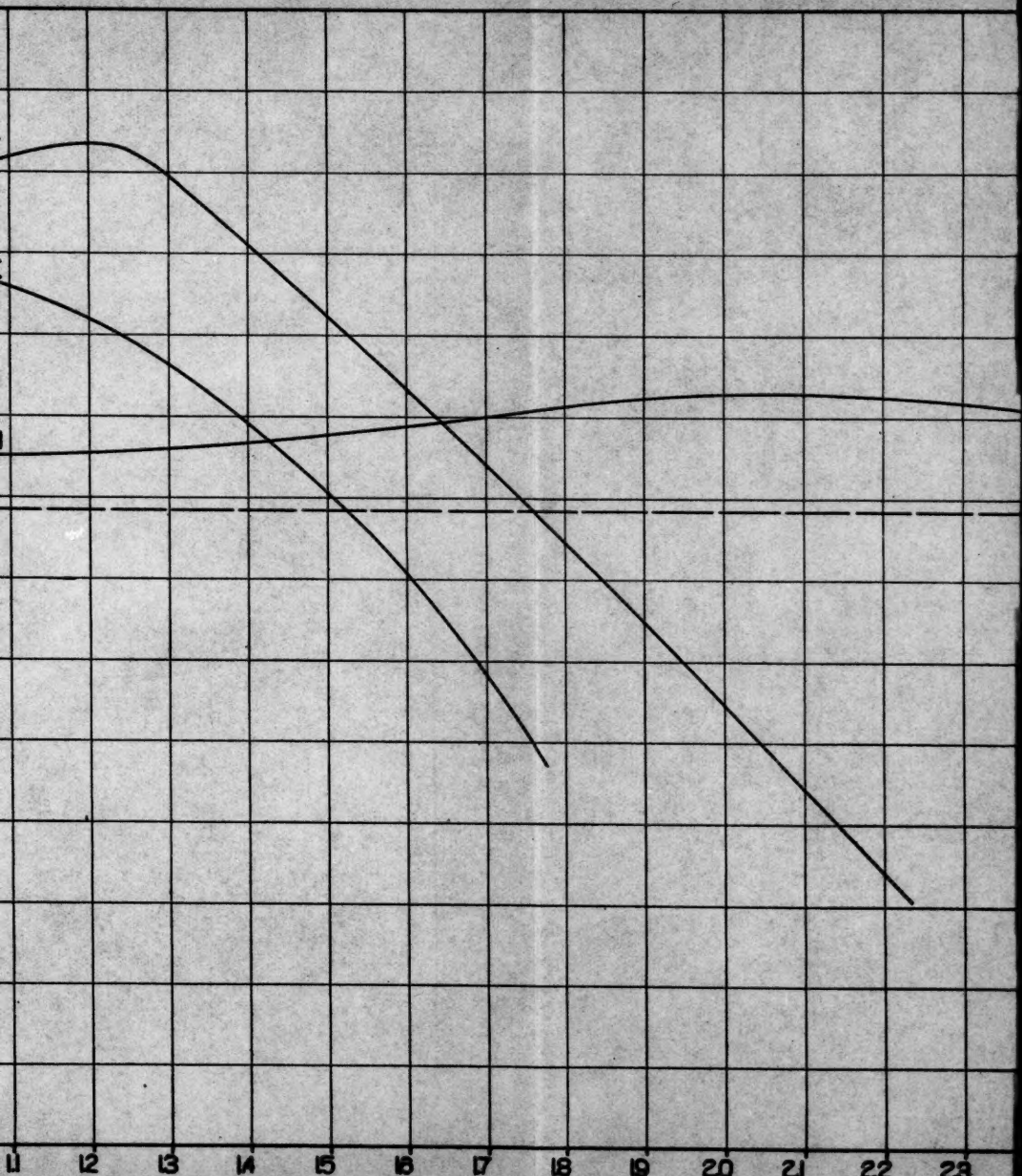


FIG. 8.—QUANTITY-HEAD



THOUSANDS OF CUBIC FEET PER MINUTE.

Y-HEAD CHARACTERISTICS OF UNIT FANS.

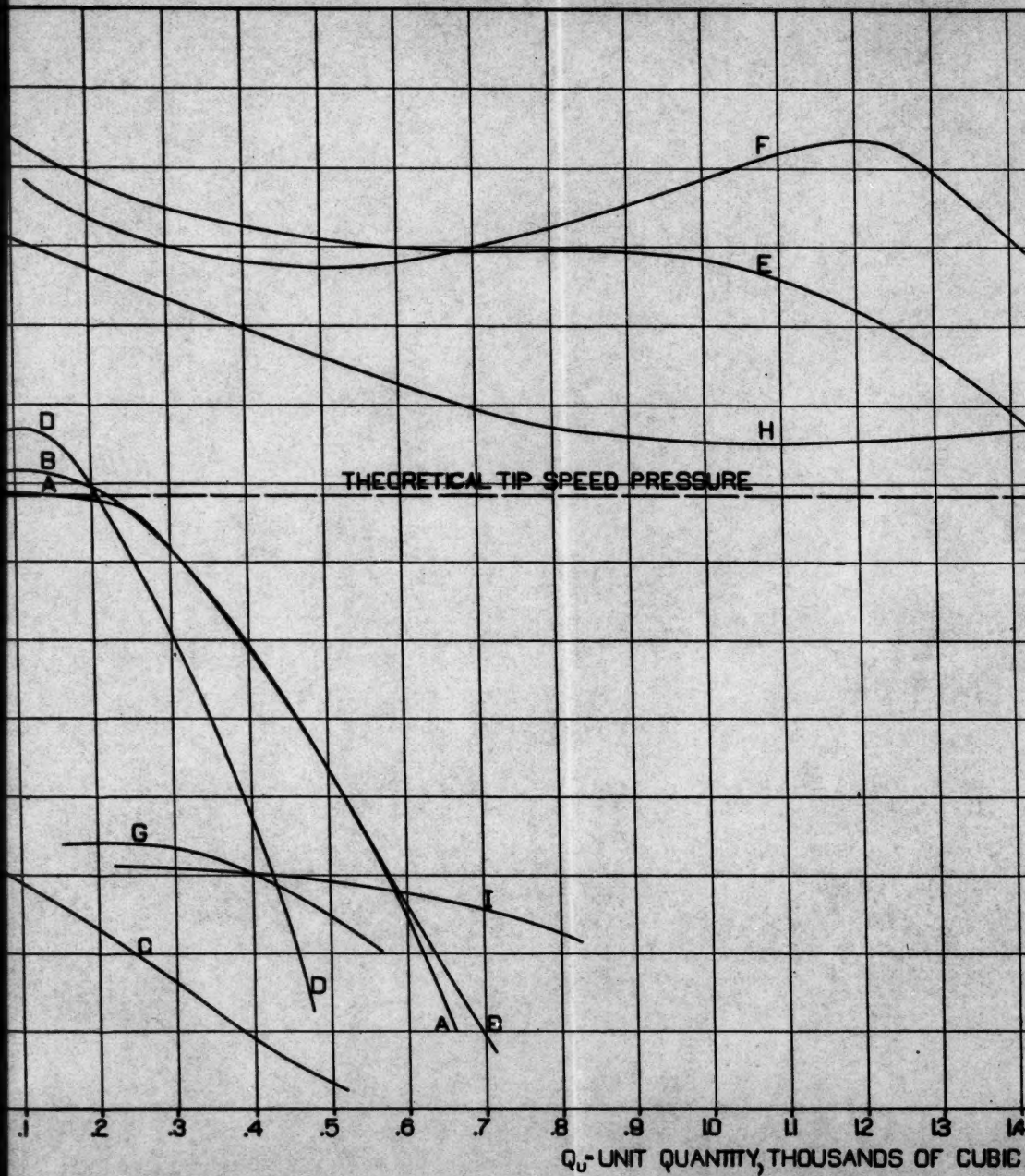
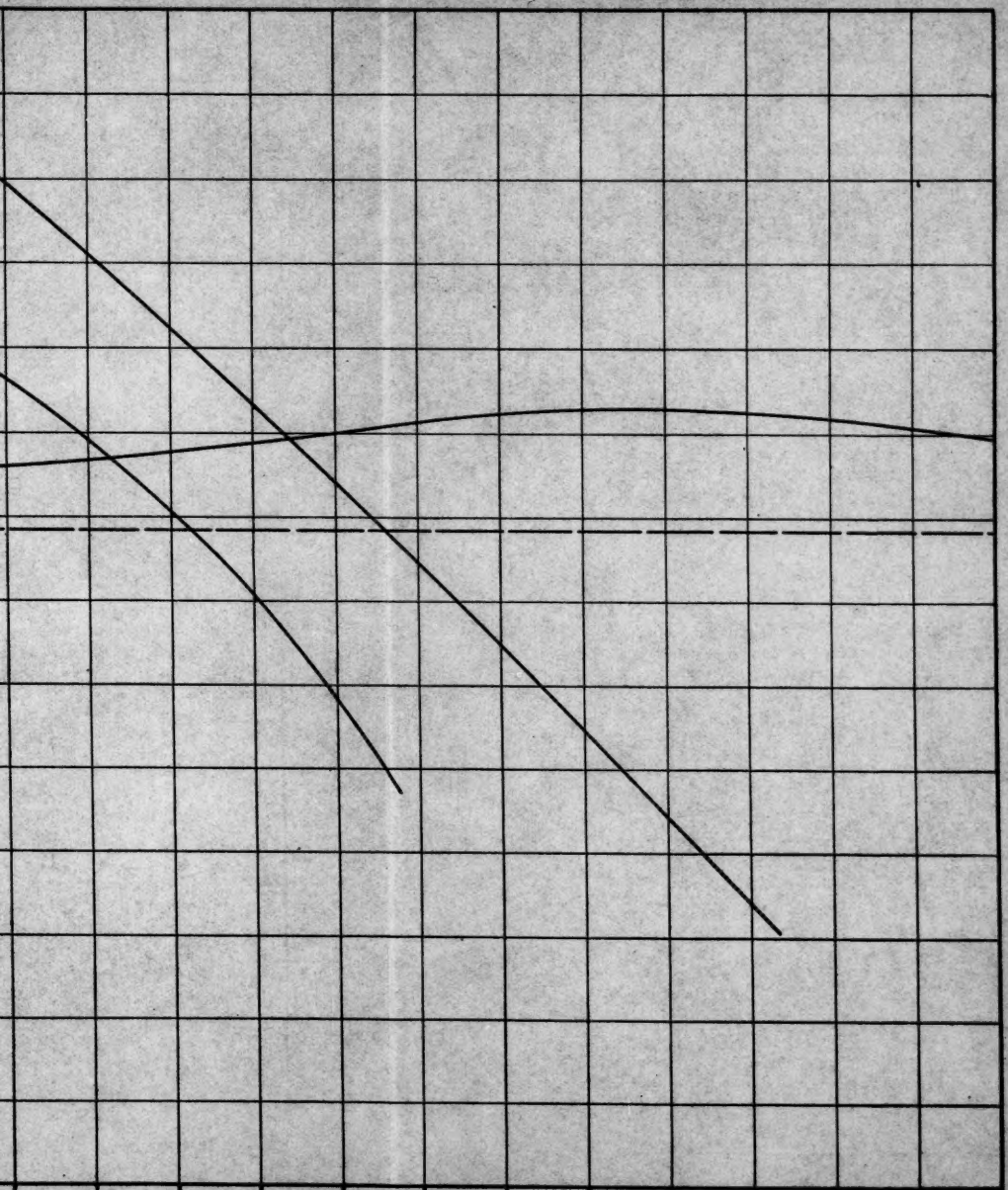


FIG. 8.—QUANTITY-HEAD CHARACTERISTI



F CUBIC FEET PER MINUTE.

ACTERISTICS OF UNIT FANS.

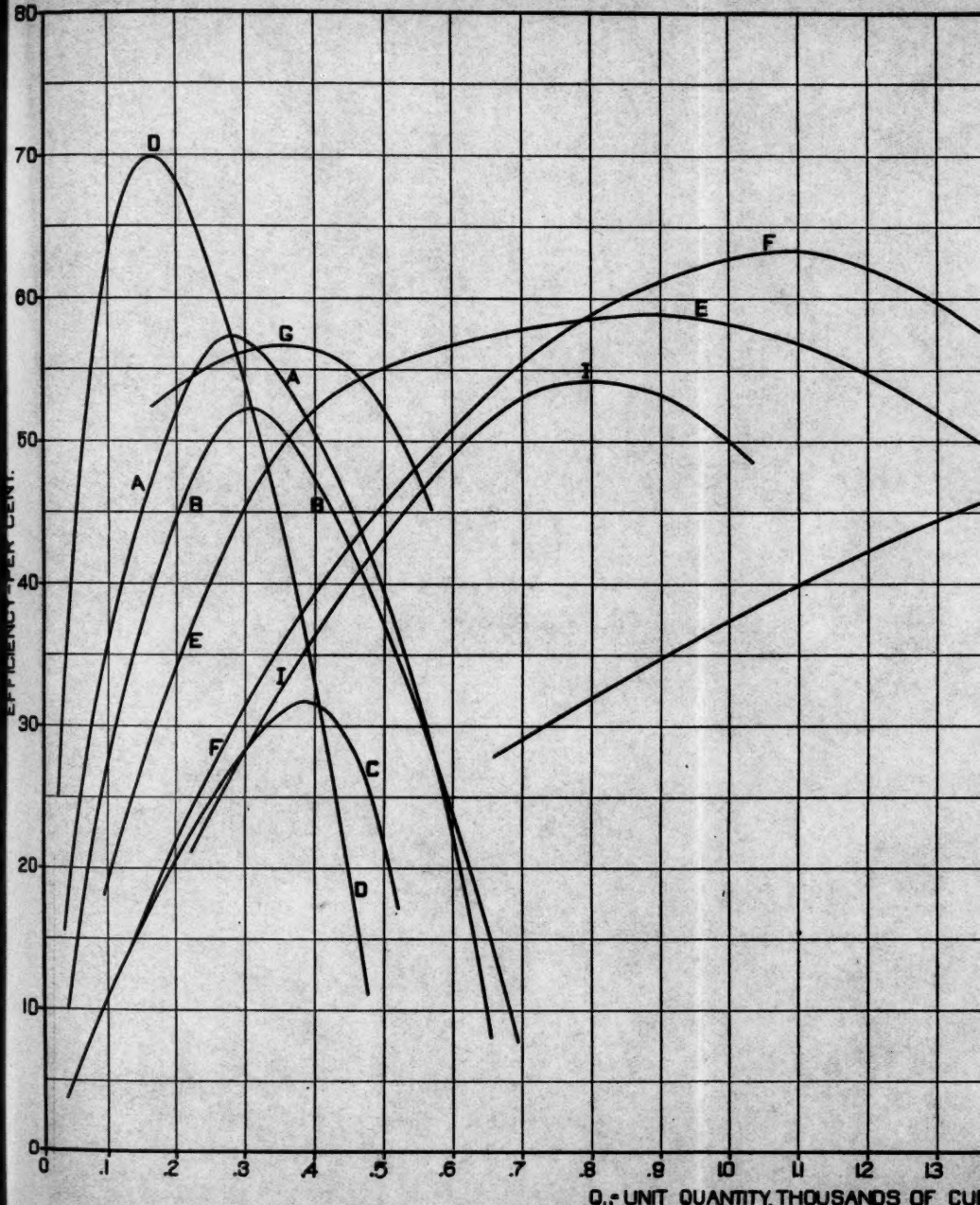
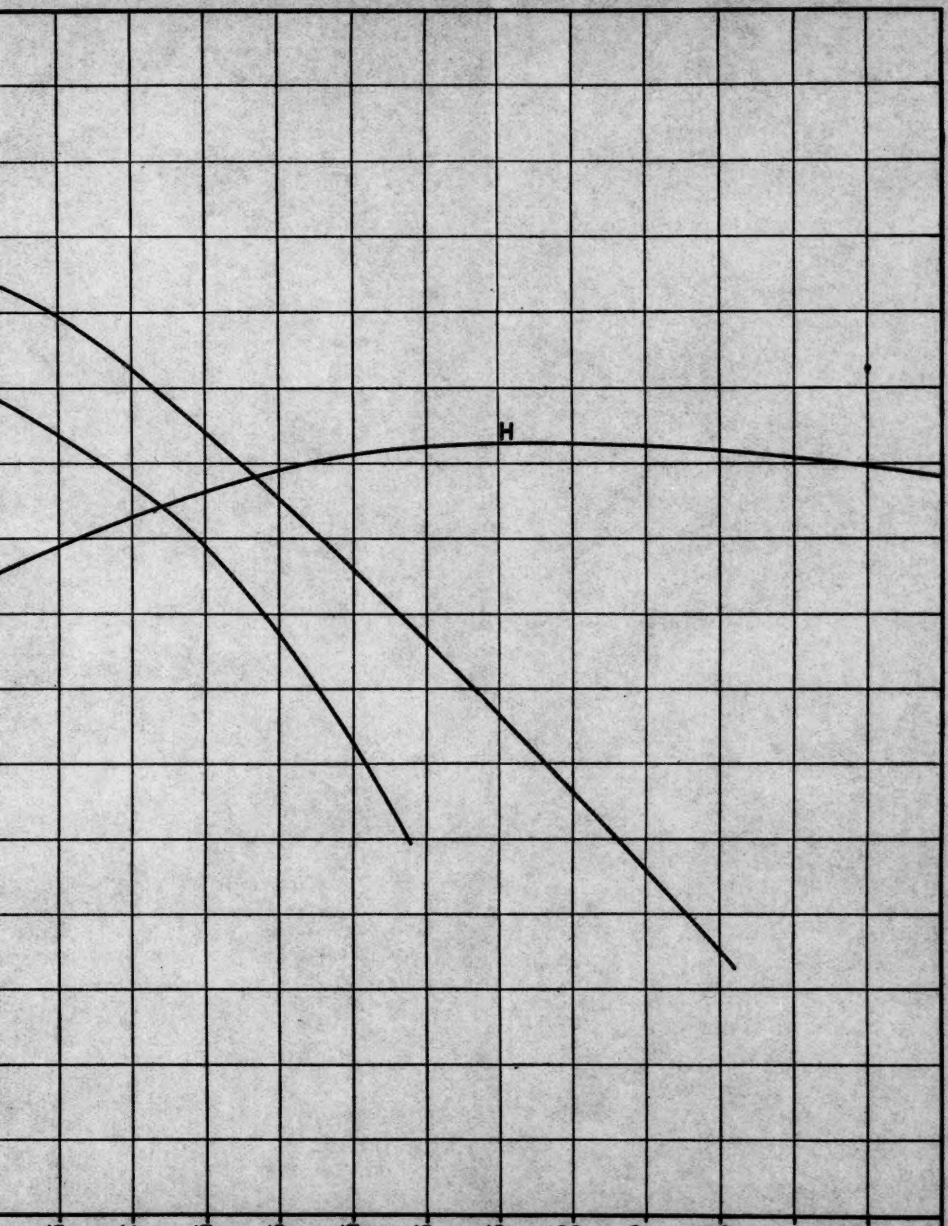


FIG. 9.—QUANTITY-EFFICIENCY CU



DS OF CUBIC FEET PER MINUTE.

CIENCY CURVES OF UNIT FAN.

values of V_s and the corresponding values of tip speed, as indicated by these curves, in order to obtain a good efficiency.

Fig. 6 shows that for given values of quantity and head some types of fans have a constant value of tip speed over a wide range of diameters, and that for all types of fans the tip speed increases rapidly as the size of the fan is decreased.

Fig. 7 shows that for most types of fans there is a range in speed (rev. per minute) over which the tip speed is constant, and that for higher speeds the tip speed rapidly increases.

Fig. 8 contains the usual quantity head characteristics for all the types, reduced to a common size of one foot diameter and 1,000 revolutions per minute. The well known characteristics of the multiblade, plate, cone and propeller types of fans may be recognized. The line of theoretical tip speed pressure is drawn in to show to what extent the pressure of the various fan types reaches the value of h corresponding to $\frac{V^2}{2g}$. The curves of Fig. 8 may be used to compare characteristics, or in making a selection of type, but are not well adapted to the solution of problems.

Fig. 9 shows that the quantity-efficiency characteristics of all fans are of a quite similar shape. These curves may be of use in the selection of a type of fan, but are not adapted to the solution of problems.

SOLUTION OF PROBLEMS.

In problems concerning the selection and performance of fans, the usual factors are: type of fan, size or diameter, speed, quantity, head, efficiency, and power. In any problem a certain number, usually four, of these factors are known or assumed, and the remainder determined. The use of the formulae and curves of this paper in solving problems in fan engineering will be illustrated by means of working out several definite examples.

Example I.

Given: Type of fan = D;
Diameter = 30 inches;
Quantity = 6,000 cubic feet per minute;
Head = 10 inches.

Determine: Speed, efficiency and power required. As the diameter, head and quantity are known, the specific diameter may be computed from formula (15), $D_s = D\frac{h^{\frac{1}{2}}}{Q^{\frac{1}{2}}}$. Then

$$D_s = 30\frac{\overline{10}^{\frac{1}{2}}}{\sqrt{6}} = \frac{30 \times 1.78}{2.45} = 21.78 \text{ inches.}$$

From Fig. 4,
for $D_s = 21.78$, it is found that
$N_s = 660$; then from formulae (11),

$$N = N_s\frac{h^{\frac{1}{2}}}{Q^{\frac{1}{2}}}, \text{ we have } N = 660\frac{\overline{10}^{\frac{1}{2}}}{\sqrt{6}}$$

$$N = \frac{660 \times 5.63}{2.45} = 1,515 \text{ revs. per minute.}$$

From Fig. 3, for $D_s = 21.78$, it is found that efficiency = 61.8 per cent., then the power required,

$$P = \frac{0.1573Qh}{e} = \frac{0.1573 \times 10 \times 6}{0.618} = 15.28 \text{ horsepower.}$$

Example II.

Given: Type of fan = A;
Diameter = 84 inches;
Speed = 360 revolutions per minute;
Head = 4 inches.

Determine: Quantity, efficiency and power. Tip speed, $V = \frac{\pi DN}{12} = \frac{\pi \times 84 \times 360}{12} = 7,920$ feet per min. Specific tip speed, $V_s = \frac{V}{\sqrt{h}} = \frac{7,920}{2} = 3,960$.

From Fig. 6, for $V_s = 3{,}960$, $D_s = 17.5$.

From equation (15), $D_s = D\frac{h^{\frac{1}{2}}}{Q^{\frac{1}{2}}}$, then

$$Q^{\frac{1}{2}} = \frac{D}{D_s}h^{\frac{1}{2}}, \text{ and } Q = \left(\frac{D}{D_s}\right)^2 h^{\frac{1}{2}};$$

substituting values,

$$Q = \left(\frac{84}{17.5}\right)^2 \sqrt{4} = 4.8^2 \times 2 = 46.$$

Quantity = 46,000 feet per minute.

From Fig. 5, for $V_s = 3{,}960$, eff. = 0.530 = 53 per cent.

Horsepower required $= \frac{0.1573 \times 4 \times 46}{0.530} = 54.5$ horsepower.

These results may be checked thus:

from Fig. 7, for $V_s = 3{,}960$, $N_s = 870$;
from Fig. 2, for $N_s = 870$, eff. = 0.530;
from Fig. 4, for $N_s = 870$, $D_s = 17.5$;
from Fig. 3, for $D_s = 17.5$, eff. = 0.530.

If it is desired to see at what point on the characteristic curve the fan will be operating for these conditions, this information may be obtained from the curves for fan A on Figs. 8 and 9,

from Fig. 9, for 53 per cent. efficiency, $Q_u = 0.375$, and
from Fig. 8, for $Q_u = 0.375$, $h_u = 0.625$.

This is a good operating point for a plate fan, and is at about the point on the characteristics at which steel-plate fans are rated.

Example III.

Given: Type of fan = G;
Speed = 1,500 rev. per minute;
Quantity = 30,000 cubic feet per minute;
Head = 6 inches.

Determine: Diameter, efficiency and power.

From equation (12), $N_s = N\frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}} = 1{,}500\frac{\sqrt{30}}{6^{\frac{3}{4}}} = \frac{1{,}500 \times 5.477}{3.833}$

$= 2{,}142$ rev. per minute.

From Fig. 4, for a value of $N_s = 2{,}142$, $D_s = 11.6$;

Then, from equation (14), $D = D_s\frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}}$

$$D = 11.6 \times \frac{\sqrt{30}}{6^{\frac{3}{4}}} = 11.6 \times \frac{5.477}{1.568} = \underline{40.5} \text{ inches.}$$

From Fig. 2, for $N_s = 2{,}142$, efficiency $= 0.505$.

$$\text{Power } \frac{0.1573 \times 6 \times 30}{0.505} = 56.0 \text{ horsepower.}$$

A 40½-inch diameter fan would probably require a special construction. If 42 inches is the next nearest stock size of this type, the cost of this would probably be less than the construction of a special fan. Assuming a 42-inch diameter fan, solve for speed required to meet the given conditions of quantity and head.

$$D_s = D\frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}} = 42 \times \frac{1.568}{5.477} = 12.03;$$

From Fig. 4, $N_s = 2{,}005$, and $N = N_s\frac{h^{\frac{3}{4}}}{Q^{\frac{1}{2}}}$;

$$N = 2{,}005\frac{3.833}{5.477} = \underline{1{,}405} \text{ revolutions per minute.}$$

From Fig. 2, for $N_s = 2{,}005$, efficiency $= 52.3$ per cent;

$$\text{Power} = \frac{0.1573 \times 6 \times 30}{0.523} = 54.2 \text{ horsepower.}$$

The 42-inch fan will require a speed of only 1,405 and a horsepower slightly smaller than that required by the 40.5-inch fan. If the speed can be reduced to 1,405 revolutions per minute, the 42-inch fan should be installed; if the speed must be maintained at 1,500 revolutions per minute a special fan with a diameter of 40.5 inches, and all other dimensions in proportion to the standard size fan, must be constructed.

Example IV.

Given: Quantity = 36,000 cubic feet per minute;
Head = 9 inches;
Speed = 1,200 revolutions per minute.
Eff. = at least 40 per cent.

Determine: Type of fan to be used, diameter and power required.

The quantity, head and speed being known, the specific speed may be computed from equation (12) $N_s = N\frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}} = 1{,}200\frac{\sqrt{36}}{9^{\frac{3}{4}}} = \frac{1{,}200 \times 6}{5.20} = 1{,}385$. A fan with this value of N_s must be selected. From Fig. 2, at $N_s = 1{,}385$, it is seen that fans G, H, E and F are available, for these have efficiencies above the lower limiting value of 40 per cent. These fans will be compared by the tabulation following:

Fan.	G	H	E	F
Type.	Cone, single width, backward tipped.	Cone, double width, forward tipped.	Single width multiblade.	Single width multiblade.
Speed, r.p.m.	1,200	1,200	1,200	1,200
N_s	1,385	1,385	1,385	1,385
D_s	15.32	9.2	9.55	9.1
Diameter, inches	53.2	31.8	33.0	31.5
Efficiency	0.56	0.49	0.46	0.45
Horsepower	91.0	104.0	111.0	113.2
Tip speed, feet per min.	16,700	10,000	10,700	9,900

On a basis of best efficiency G, of course, would be selected. This fan, however, is considerably larger and would cost much more than either of the other fans. Also, the tip speed is high, and thus calls for a more rigid construction. Between fans H, E and F, there is not much choice on a basis of diameter, but fan H is the most efficient of the three, although, being double width, would weigh and cost more than either

E or F. Between fans E and F the selection would probably be type F, because it is smaller, has a lower tip speed, and requires only slightly more power than fan E. The conclusion is that fan G should be selected if the basis is efficiency only, and fan F should be selected on a basis of first cost and size.

Other types of problems arising in fan engineering may be solved by the use of the curves and formulae, examples of which cannot be given on account of lack of space; for instance, the diameter, quantity, head and efficiency may be given and the type and speed determined; or the type, diameter, quantity and speed may be given and the head, efficiency and power determined.

The choice of type of fan is often made upon a basis of the shape of the capacity head characteristic. For instance, in heating and ventilating work, or in forced draft for oil-burning installations, it may be desirable to have a constant pressure over a wide range of volume at constant fan speed. This would call for a "flat characteristic" of the type shown by E, F or H, Fig. 8. In mechanical-stoker on gas-producer work a "steep characteristic" is desired, such as would be obtained from fans of type A, B or D. In these cases, only those types having the desired characteristics need to be considered in selecting the type, size and speed of fan required to meet the conditions of a given installation.

Choice of type may also be based on the amount and character of the noise produced, and this is an important factor in hotel, office and theater installations, and recently has received consideration in the selection of the forced-draft fans for destroyers in the naval service.

The shape of the quantity-power characteristic should also receive consideration in the selection of the type of fan. In certain installations it is important that the fan should not overload the motor if the resistance to the flow of air is removed and thus result in a large increase in the quantity of air discharged. This calls for a power characteristic at constant speed which reaches a maximum near the rated quantity, and falls off for larger quantities.

CONSTRUCTION OF CONSTANT SPEED CHARACTERISTICS.

When the type, diameter and speed of fan have been selected for a certain installation, it is desirable to have complete characteristic curves of capacity *vs.* head, efficiency and power for the speed at which the fan is to operate. Data for these characteristic curves may be obtained most readily from the unit-fan curves. Simultaneous values of Q_u, h_u and efficiency are selected from the unit-fan curves for the given type of fan on Figs. 8 and 9, and the corresponding values of Q, h and P are computed from equations (21), (22) and (23).

As an example of the method of plotting these curves, the computations will be made for the fan computed in problem I. This is a 30-inch diameter fan of type D running at 1,515 revolutions per minute. The source of the items of Table II, by means of which the computations are made, are as follows:

Item 1. = Quantity discharged by unit fan, thousands of cubic feet per minute, selected from curve for type D, Fig. 8.

Item 2. = Static efficiency, from curve of type D, Fig. 9, corresponding to item 1.

Item 3. = Static head for unit fan, inches of water, from curve of type D, Fig. 8, corresponding to item 1.

Item 4. = Quantity, actual fan, thousands of cubic feet per minute, from formulae (21).

$Q = Q_u n d^3$, in which

Q_u = item 1; $n = \frac{1,515}{1,000} = 1.515$,

$d = \frac{30}{12} = 2.5$; $d^3 = 15.625$,

or Q = item 1 times 23.68.

Item 5. = Static head, actual fan, inches of water, from formula (22), $h = h_u n^2 d^2$, in which

h_u = item 3, $n^2 = \overline{1.515}^2 = 2.295$,

$d^2 = \overline{2.5}^2 = 6.25$; then h = item 3 times 14.33.

Item 6. = Horsepower required, actual fan, from formulae (16);

$$P = \frac{0.1573\, Qh}{e} = \frac{0.1573 \times \text{item 4} \times \text{item 5}}{\text{item 2}}.$$

TABLE II.—COMPUTATIONS FOR CONSTANT SPEED CHARACTERISTICS. FAN TYPE D; DIAMETER = 30 INCHES. SPEED = 1,515 R.P.M.

Item.	Q_u 1	e 2	h_u 3	Q 4	h 5	P 6
1	0.05	0.43	0.850	1.182	12.18	5.26
2	0.10	0.62	0.870	2.368	12.47	7.50
3	0.15	0.694	0.852	3.550	12.20	9.83
4	0.17	0.70	0.836	4.025	11.97	10.81
5	0.20	0.685	0.792	4.735	11.34	12.60
6	0.25	0.623	0.705	5.915	10.10	15.10
7	0.30	0.545	0.605	7.100	8.68	17.76
8	0.35	0.45	0.495	8.280	7.10	20.55
9	0.40	0.335	0.372	9.47	5.33	23.70
10	0.45	0.20	0.230	10.64	3.30	27.60

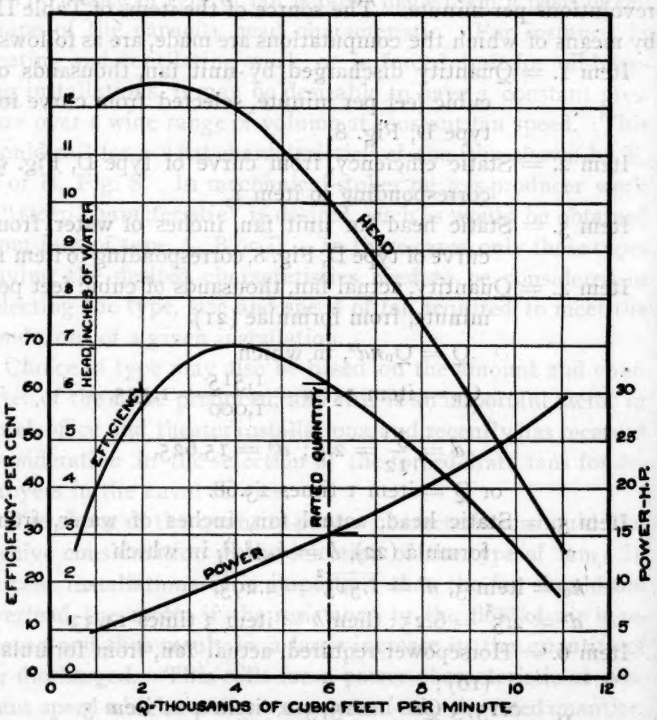


FIG. 10.—CONSTANT SPEED CHARACTERISTICS OF 30-INCH FAN, TYPE D, SPEED, 1,515 R.P.M. PLOTTED FROM TABLE II.

The values of Q, e, h and P thus obtained, when plotted as in Fig. 10, form the usual characteristics covering the performance of the fan at constant speed.

Data for curves at other speeds may be computed by means of an extension Table II, and a group of curves for various speeds thus plotted on the coordinates of Fig. 10, which will present the complete performance of the fan.

CONSTRUCTION OF CONSTANT HEAD CHARACTERISTICS.

Another type of characteristic for a given fan which is sometimes desired is that showing the fan performance at constant static head. For instance, in forced-draft installations the quantity requirements may vary over a wide range, depending upon the number of boilers in service, and at the same time the pressure required in the fireroom, or under the grates, should remain constant. In this case it is desired to know the variation in speed and power required to produce the various quantities at constant head.

TABLE III.—COMPUTATIONS FOR CONSTANT HEAD CHARACTERISTICS. FAN TYPE D; DIAMETER = 30 INCHES. HEAD = 10 INCHES.

	Q	Q^1	D_s	N_s	N	e	P	V
Item.	1	2	3	4	5	6	7	8
1	2.0	1.414	37.70	350	1,393	0.610	5.16	10,950
2	4.0	2.000	26.70	500	1,410	0.699	9.00	11,070
3	6.0	2.449	21.78	660	1,515	0.618	15.28	12,000
4	8.0	2.828	18.85	830	1,650	0.520	24.20	12,970
5	10.0	3.162	16.85	1,020	1,815	0.432	36.40	14,270
6	12.0	3.464	15.40	1,220	1,980	0.360	52.50	15,550

Results plotted on Fig. 11.

As an illustration, computation will be made for the characteristics of the fan of problem I, when operating against a constant static head of 10 inches. The computations may be most easily made by means of a tabulation, as given in Table III, the source of the items of which is as follows:

Item 1. = Q, quantity, thousands of cubic feet per minute, assumed values.

Item 2. = $\overline{\text{item 1}}^{\frac{1}{4}}$.

Item 3. = D_s, from formula (15); $D_s = D\frac{h^{\frac{1}{4}}}{Q^{\frac{1}{4}}}$;

$$D_s = \frac{30 \times \overline{10}^{\frac{1}{4}}}{Q^{\frac{1}{4}}} = \frac{53.37}{\text{Item 2}}.$$

Item 4. = N_s, corresponding to item 3 from curve for fan type D on Fig. 4.

Item 5. = N, speed of fan, rev. per minute from formula (11);

$$N = N_s\frac{h^{\frac{1}{4}}}{Q^{\frac{1}{4}}};\ h^{\frac{1}{4}} = \overline{10}^{\frac{1}{4}} = 5.63 :\ N = \frac{\text{item 4} \times 5.63}{\text{item 2}}.$$

Item 6. = Efficiency, corresponding to D_s, item 3, from curve for fan type D, on Fig. 3.

Item 7. = Horsepower, from formula (16),

$$HP = \frac{0.1573\ Qh}{e} = \frac{0.1573 \times \text{item 1} \times 10}{\text{item 6.}}.$$

Item 8. = Tip speed of fan, feet per minute, from equation,

$$V = \frac{\pi DN}{12} = \frac{\pi \times 30 \times \text{item 5}}{12}.$$

The values of Q, P, e, N and V thus obtained may be plotted as shown in Fig. 11 to form the characteristics of the fan at a constant head of 10 inches. If the maximum safe tip speed for this type of fan is 14,000 feet per minute, the maximum quantity obtainable, and the corresponding speed, efficiency and power for 10 inches static pressure may be obtained as shown on Fig. 11.

Data for the constant speed characteristics for any size fan of a type may also be computed from the curves of specific speed and specific diameter of that type. Values of N_s and D_s corresponding to certain efficiencies are selected from the specific fan curves, and the desired data for the characteristics computed by means of equations (12), (15), and (16).

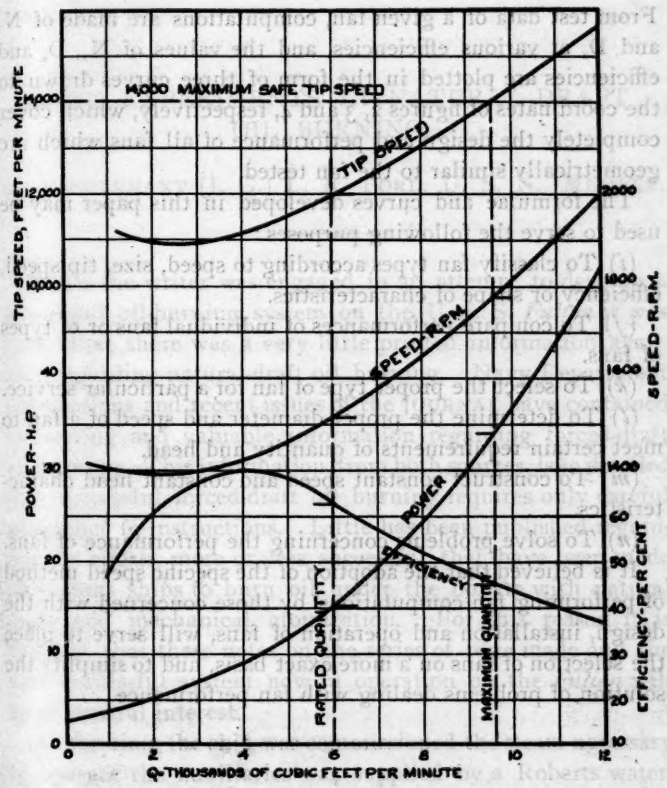


FIG. 11.—CONSTANT HEAD CHARACTERISTICS OF 30-INCH FAN, TYPE D, HEAD = 10 INCHES. PLOTTED FROM TABLE III.

SUMMARY.

The specific speed and the specific diameter of a fan operating at any given efficiency are defined as the speed and diameter, respectively, of a geometrically similar fan which will deliver one thousand cubic feet of air per minute against one inch static head at the same efficiency. The formulae for specific speed, $N_s = N \frac{Q^{\frac{1}{2}}}{h^{\frac{3}{4}}}$, and for specific diameter, $D_s = D \frac{h^{\frac{1}{4}}}{Q^{\frac{1}{2}}}$, are derived from the known laws of fan performance.

From test data of a given fan, computations are made of N_s and D_s at various efficiencies, and the values of N_s, D_s and efficiencies are plotted in the form of three curves drawn to the coördinates of figures 2, 3 and 4, respectively, which cover completely the design and performance of all fans which are geometrically similar to the fan tested.

The formulae and curves developed in this paper may be used to serve the following purposes:

(*i*) To classify fan types according to speed, size, tip speed, efficiency or shape of characteristics.

(*j*) To compare performances of individual fans or of types of fans.

(*k*) To select the proper type of fan for a particular service.

(*l*) To determine the proper diameter and speed of a fan to meet certain requirements of quantity and head.

(*m*) To construct constant speed and constant head characteristics.

(*n*) To solve problems concerning the performance of fans.

It is believed that the adoption of the specific speed method of performing fan computations by those concerned with the design, installation and operation of fans, will serve to place the selection of fans on a more exact basis, and to simplify the solution of problems dealing with fan performance.

SOME EXPERIMENTS IN NATURAL-DRAFT OIL BURNING.

By LIEUTENANT (J. G.) L. R. FORD, U. S. N., MEMBER.

While the writer was engaged in an attempt to develop a successful oil-burning system on the U. S. S. *Fulton* it was found that there was a very little printed information available regarding natural-draft oil burning. Navy Department publications and recent issues of the JOURNAL have contained interesting and valuable information regarding forced-draft oil burning. This information, from both sources, is so detailed that successful forced-draft oil burning requires only careful adherence to instructions. Little has been published regarding the efforts, more or less successful, that have been made on various ships to burn oil under the boilers with natural draft and mechanical atomization. For this reason it is thought that these notes on the series of tests made and the very successful system now in operation on the *Fulton* will be of general interest.

At the time the ship was commissioned the steam necessary to operate the auxiliaries was supplied by a Roberts water-tube boiler equipped with one large burner which used either steam or air as the atomizing agent. The air for combustion was supplied by natural draft through two small ventilators and through the door between engine room and fireroom. After a disastrous furnace explosion, due to the burner flame being extinguished and the furnace becoming filled with explosive gas, which ignited, the Bureau ordered this burner removed. The boiler was then fitted with two of the Bureau's Standard, Type B, air registers and Type I burners. These burners atomize the oil mechanically and are designed for use with forced draft.

During the last overhaul period at the navy yard extensive alterations were made in the machinery installation in accordance with the recommendations of the Trial Board, including the installation of an additional boiler, this boiler being of the Almy water-tube type. The installation of this boiler necessitated a complete rearrangement of all machinery and piping in the fireroom and the moving of the old boiler to the side of the fireroom, to make room for the new one. This rearrangement resulted in a very crowded condition and curtailed the already limited air supply.

As soon as the ship was ready to leave the navy yard the burners were lighted under one boiler and it was at once evident that the combustion was very poor. The incomplete combustion resulted in the formation of dense black smoke and caused a thick, heavy soot and an asphaltum residue to deposit on all the heating surfaces and on the inside of the casing and smokepipe. The bad furnace conditions were aggravated by the forcing required to meet the extra demand for steam, caused by severe weather conditions. More trouble was experienced with the new boiler than with the old one, due to several reasons: This boiler was farther from the fireroom door, and the connection between uptake and smokepipe very indirect, consequently the draft was poor. In addition to this the space between tubes in this boiler is very small, so that the soot quickly clogged the tubes. For these reasons the Roberts boiler was depended upon to supply steam, with a little help occasionally from the Almy boiler.

While various schemes for improving the furnace conditions were being tried the condition of the Roberts boiler became rapidly worse. The construction of this boiler is such that the main generating tubes cannot be cleaned on the fire side without putting the boiler out of use. As a result the soot and residue rapidly collected, greatly reducing the draft and the evaporative capacity.

Small, portable, electric blowers were placed on the fireroom-floor plates so as to discharge against the registers in the furnace front. This helped only slightly. A sheet-metal

box was built on the boiler front so as to completely enclose the registers and burners. This box was made airtight and connections were made for two ¼-H.P. portable blowers. This device proved to be a failure, as these blowers did not have capacity enough to supply the necessary air and they could not produce a pressure in the air box.

The air box was then removed and a smaller one constructed, that would stand on the floor plates in front of the burners. Two ¼-H.P. blowers were connected so as to discharge into this box. Two sheet-iron pipes, 2 inches in diameter, were run from this box to each register and connected to discharge through holes in the registers to the center of each burner. The air supplied in this manner was in addition to that coming from the fireroom through the register vanes, which were left wide open. This arrangement has been used on other vessels and reported as successful, but it was not found so in this case. The combustion was much improved, but far from satisfactory. So much soot was deposited that it became necessary to establish a routine requiring that the tubes be blown with steam once every hour, or 24 times each day.

The oil pressure habitually carried with the Bureau type of burner was 200 pounds. It was found that by reducing the oil pressure much better combustion was obtained, due to the increased ratio of air to oil, but the oil thus burned was insufficient to maintain the steam pressure.

An experimental burner was made by modifying one of the Bureau-type burners. A piece of 1¼-inch iron pipe was fitted around the shank of the burner, extending as far as the tip and forming an air casing around the burner. It was supplied with air from the Westinghouse compressor in the engine room. A circle of small holes was drilled in the end of the air casing so that fine jets of air were sprayed into the cone of oil spray issuing from the burner tip. This burner gave fairly good combustion, but the air jets did not enter at the correct angle. Better results would no doubt have been obtained if the air streams had been made to converge toward the center,

but this would have necessitated increasing the overall diameter of the burner so much that it could not be entered in the register, and it was not considered desirable at this time to remove the registers.

A pair of air-atomizing burners were designed and manufactured on board and installed in the Bureau-type registers. These burners were of the "inside-mixture" type—the air and oil were discharged into a mixing chamber in the tip, where the mixture was given a whirling motion before issuing from the tip. The air was obtained from the Westinghouse compressor. It was found that excellent combustion could be obtained with these burners, but very close attention on the part of the fireroom force was required to obviate the dangers due to fluctuations in the air and oil pressure, with the consequent extinguishing of the flame. This very serious defect, combined with the extravagant consumption of steam by the compressor used to supply the atomizing air, made it very desirable to continue the efforts to develop a method of using the mechanical atomization system. It would have been more economical to use steam instead of air as the atomizing agent in these burners, but in view of the poor natural air supply from the fireroom it was considered desirable to utilize the oxygen in the atomizing air to aid the combustion.

While these experiments were in progress Lieut. A. M. Penn, U. S. N., in charge of the Fuel-Oil Testing Plant, at the Navy Yard, Philadelphia, reported on board, by order of the Navy Department, to conduct experiments and make recommendations in regard to the fuel-oil burning system of the ship. To his advice and suggestions the ultimate good results obtained are largely due.

The Bureau Standard Type "B" registers were removed from the furnace front of the Roberts boiler and a bar of ¾-inch by 3-inch flat iron was supported across the front of each opening in the furnace front, on ¾-inch studs, these studs projecting 10 inches from the furnace front and being provided with threads and nuts for adjusting the bar for distance. Each

bar was drilled at its center large enough to permit the insertion of a Bureau-type I burner, a set screw through the side of the bar locking the adapter in place. A trial of the burners under this condition was made, not with any expectation of operating them in this manner, but merely to observe their action. As soon as the cone of oil spray issuing from the burner tip was ignited the flame left the burner and jumped to the back of the furnace, leaving only a whitish haze near the burner. The flame would then jump back and forth, approaching, then receding from, the burner.

A burner was then modified to conform to a burner that had been experimentally developed at the Fuel-Oil Testing Plant, at Philadelphia. This modification is shown in Fig. 1. The corners of the hexagonal tip were filed down until a

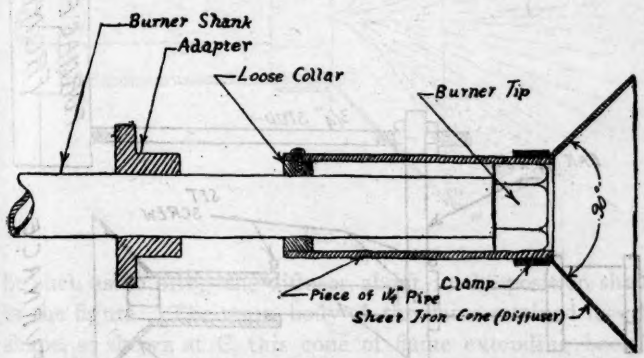


FIG. 1.

1¼-inch pipe would slip easily over the tip. A funnel-shaped "diffuser" was made of sheet iron and attached to the end of a piece of 1¼-inch pipe, 4 inches long, by means of a band clamp. This clamp method of securing the diffuser on the pipe was preferable to screws or rivets as it permitted easy removal of the diffuser. This diffuser was made with an included angle of 90 degrees and the diameter at the large end was made 4⅜ inches. During the subsequent experiments several different angles were tried, but the 90-degrees angle gave the best results. The pipe was slipped on the burner,

over the tip. A loose collar, sliding on the burner shank, supported one end, the other end being supported by the burner tip. The pipe was secured to the loose collar by a small screw. This arrangement permitted adjustment of the distance the burner tip projected into the diffuser. To remove the burner from its holder it was only necessary to take out the small screw, draw the pipe and diffuser off the end of the burner, then draw the burner out of its holder in the usual manner. This burner was mounted in a temporary holder, as shown in Fig. 2, for experimental purposes.

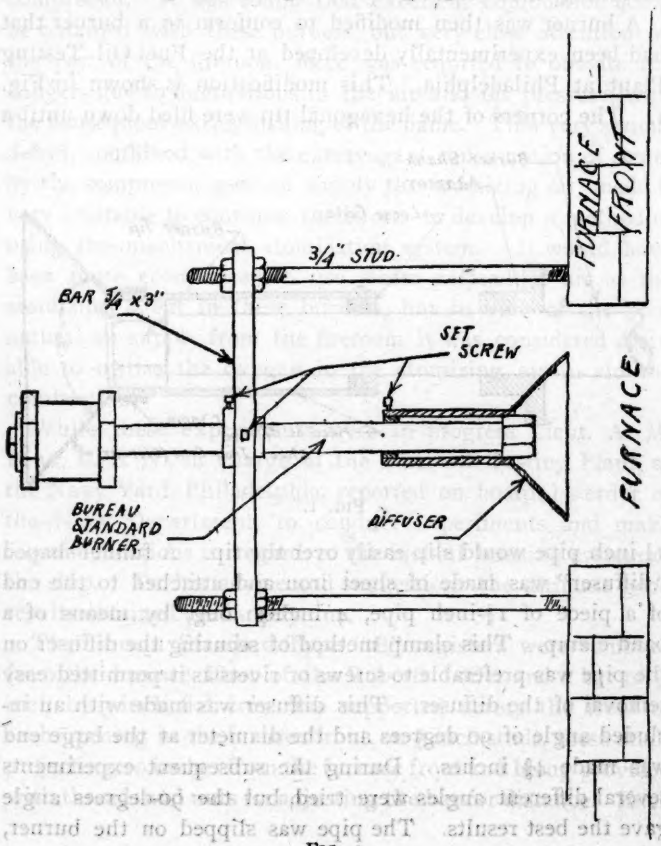


FIG. 2.

When this burner was lighted off a most interesting action was observed. With the diffuser drawn well back toward the rear the flame would jump away from the burner as previously noted, but when the diffuser was pushed forward it would pick up the flame and a short cone of clear flame would result. When the diffuser was adjusted to the best operating position a halo of white flame would appear around the edge. This action is illustrated in Fig. 3. The best position seemed to

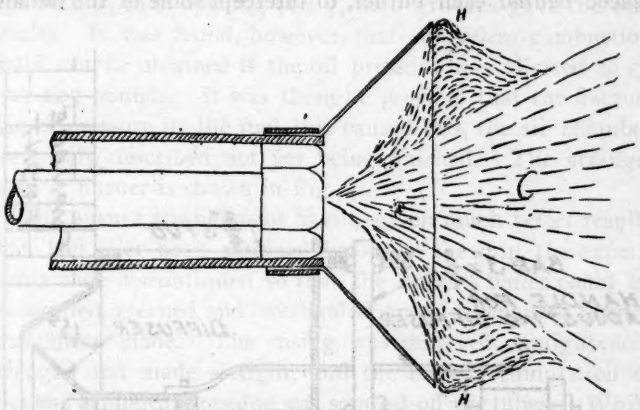


FIG. 3.

be such as to bring the diffuser about in the position shown in the figure. The main body of the flame retained its cone shape, as shown at C, this cone of flame extending back to the burner tip, but part of this flame seemed to draw out to the diffuser and curl down from the edge, as shown at H H. The combustion obtained with this arrangement was very good indeed and further development of the system seemed most desirable.

The Roberts boiler had been used for the trial of the burner, and the appearance of the flame in the furnace indicated that the vertical side tubes and the horizontal top tubes exerted a marked cooling effect on the gases and retarded combustion. To reduce this effect a checkered wall of brick was built up on each side of the furnace, against the vertical side tubes.

It was not feasible to cover the horizontal top tubes, but this would be desirable.

The next step was to fit a complete burner to each of the openings in the furnace front of the Roberts boiler. These openings were modified by fitting a cone of heavy sheet metal into each, thereby obtaining a Venturi meter effect. The openings tapered from $15\frac{5}{8}$ inches diameter at outside to 12 inches at throat. A circular baffle plate of sheet metal was placed behind each burner, to intercept some of the radiant

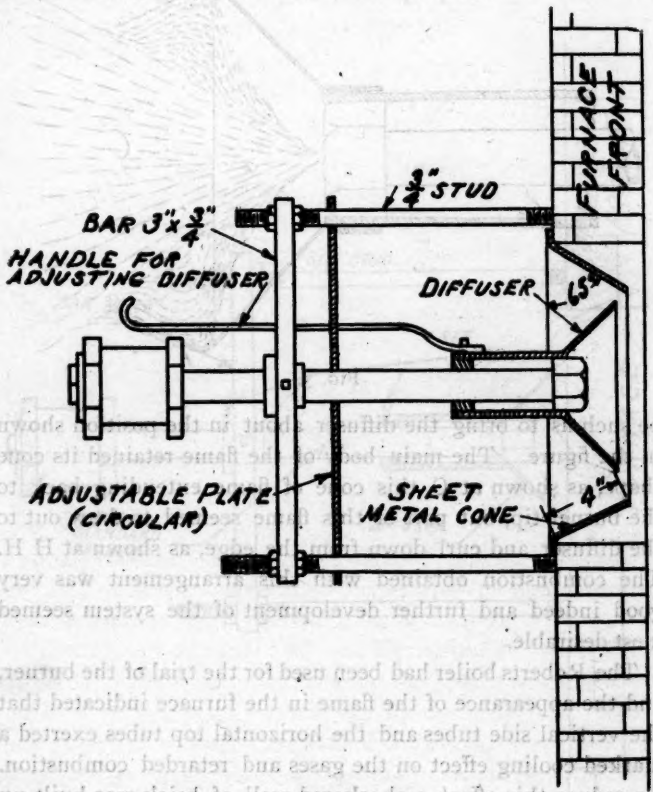


FIG. 4.

heat from the flame, and a handle was fitted for adjusting the diffuser from behind this plate. The cone-shaped openings made a very noticeable improvement in the furnace conditions. The flame obtained was short, of a dazzling whiteness, and no smoke was produced. The air flowing into the furnace by natural draft seemed to mix thoroughly with the fuel, due to the influence of the diffuser and the Venturi furnace opening. Several different sizes of burner tips were tried. The tips with $\frac{3}{64}$-inch opening and $\frac{1}{32}$-inch grooves gave the best results. It was found, however, that smokeless combustion could not be obtained if the oil pressure was allowed to go over 125 pounds. It was thought possible that the fluctuation of pressure in the fuel line caused this, the air chamber previously described not yet being finished. The arrangement of burner is shown in Fig. 4.

This burner arrangement having given much better results than had ever been obtained before on the ship, the experiments were discontinued so that the Roberts boiler could be dismantled, cleaned and overhauled and a permanent burner installation made. The casing was removed, straightened, relagged and made airtight, and the large accumulation of soot and asphaltum residue was scraped off the tubes. While this work was in progress the set of permanent burner fittings was made, arranged so that all adjustments could be easily made. The conical openings in the furnace front were modified by riveting vertical fins of thin sheet metal around the circumference, in the shape of a helix. It was thought that this would promote the mixing of air and fuel by giving the entering air a rotary motion. This assumption was later proven to be quite correct. The opportunity was taken to complete the installation of the air chamber in the dead end of the fuel line. A small pipe connection was made in the top of this air chamber to permit it being charged with air from the Westinghouse compressor.

The arrangement of burner and holder is shown in Fig. 5. With this arrangement the diffuser may be adjusted to get the best flame; the burner may be moved in or out of the open-

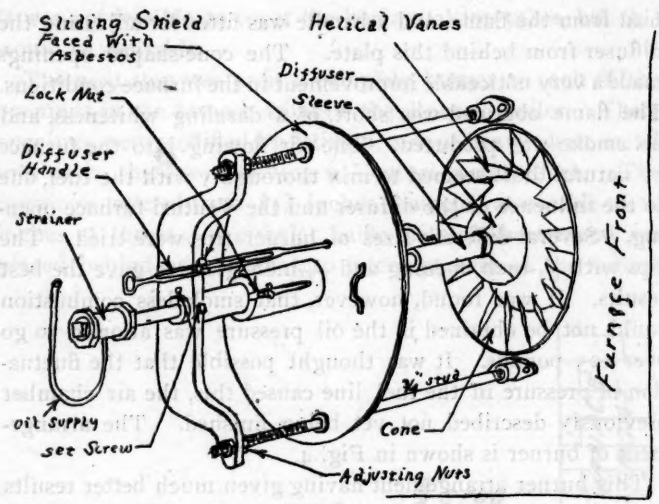


FIG. 5.

ing in the furnace front; the circular baffle plate may be moved in or out to regulate the air supply, or, if necessary, close the opening altogether; and the whole burner with all of its fittings may be moved in or out by means of the nuts on the studs. Each adjustment is independent of all the others. The sockets into which the studs are screwed may be adjusted, by liners, to throw the burner to one side or the other.

When the overhaul and cleaning of the boiler was completed it was reassembled, the large air chamber was mounted on the dead end of the oil line and the new burner holders and burners installed on the furnace front, using burners with $\frac{3}{64}$-inch tips and $\frac{1}{32}$-inch plugs.

When the burners were lighted the results were most gratifying. The flame from each burner was very short and dazzlingly white and the gas in the furnace was quite clear. As soon as the brick walls on the sides of the furnace heated up all signs of smoke disappeared. The influence of the helical vanes on the air was quite marked. The large air chamber

kept the oil pressure steady, the gage hand remaining perfectly stationary at any point desired. The effect of this steady oil pressure was quickly noted at the burners. When the diffusers were adjusted to get the halo around the edges it remained there without any further adjustment. There was an entire absence of smoke at the top of the smokepipe and the burner steamed rapidly. The most surprising result was the low oil pressure required to maintain the steam pressure and the wide range of pressure through which the burners could be operated smokelessly. As previously mentioned, it had been necessary under the old conditions to operate the burners under an oil pessure of 200 pounds. It was now found that the steam necessary to meet the normal demand of the ship could be evaporated with an oil pressure of 50 to 100 pounds, while at times, when an extra demand for steam was made, the burners operated smokelessly up to 200 pounds. From the standpoint of evaporative efficiency absolute smokelessness is not the most desirable condition; but it is just the condition most desirable for this boiler, as it prevents fouling the tubes, which are not accessible for cleaning.

After a little experimenting it was found that the steam pressure could be quickly and closely regulated by varying the oil pressure. This could be done by varying the spring tension of the regulator valve on the oil pump. Steam control was thus reduced to the mere screwing of a nut up or down with a wrench. Needless to say, this novelty appeals to the firemen. The pressure variation required to meet the ordinary changes of load were found to be so small that no further adjustment of the diffusers was required after they had once been properly set, but any large change called for a slight readjustment.

A glance at the curves of discharge for the Bureau standard burners, published in the Bureau's Instructions for Oil Burning, will show what the great reduction in oil pressure meant in the line of oil saving. It was estimated that under the new conditions about half as much oil was burned as had been burned before the burners were improved. This

estimate was based on the above-mentioned curves, but in order to check the estimate, the oil consumption was measured accurately for two days by means of two small lubricating oil tanks which were calibrated and connected to the fuel-feed pump suction. This test showed the daily oil consumption to be 500 to 600 gallons. The consumption had previously been 1,000 to 1,200 gallons.

It was thought that the diffusers, the cones in the furnace front and the vanes on the cones, all being of light sheet iron, would quickly burn out, and for this reason a number of spares were made. The burners have, however, at date of writing, been in constant operation for over one month and the above-mentioned parts show no signs of burning, except one vane on which the flame impinges, due to the burner not being exactly centered. The cold air rushing in from the fireroom seems to keep the parts cool.

The helical vanes, in addition to giving the air a rotary motion and thereby promoting mixture of air and oil, serve a useful purpose in intercepting the radiant heat from the flame. It is found that since they were installed the heat in front of the boiler is no greater than when the closed registers were in use.

With reference to the regulation of the steam by varying the oil pressure, it should be noted that past experience has shown this method to be incorrect in the case of the standard forced-draft installations, where any change in oil pressure changes the relative velocity of air to oil. In the case of this particular installation, this method of regulation has proven highly successful, and is much more satisfactory than regulating by changing the number of burners in use. The writer does not recommend it for every installation; each one should be regulated by the method found best in practice.

While the calibrated tanks were being used to measure the oil consumption it was found that for a time the rate of discharge did not correspond to that shown on the Bureau curves for the pressure then carried. The small strainers fitted to the end of each burner were opened up and it was found that

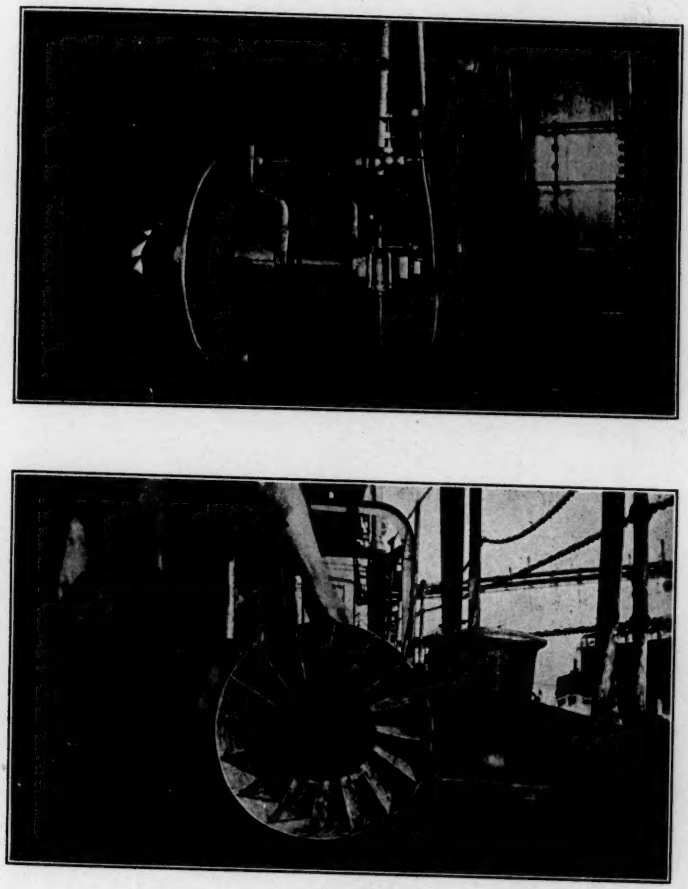


FIG. 6.

the wire gauze was badly clogged. After the strainers were cleaned the rate of discharge corresponded to the pressure.

The photograph in Fig. 6 shows the general arrangement of the two burners and holders as installed, the second burner being alongside the one in the foreground. The lower photograph shows very clearly the construction of the sheet-metal cone and the helical veins.

U. S. S. *TUCKER.*

DESCRIPTION.

By Lieutenant Ormond L. Cox, U. S. Navy, Member.

Torpedo-Boat Destroyer No. 57, the *Tucker,* is one of the six destroyers of the same class authorized by an Act of Congress approved March 4, 1913. The *Tucker* is a twin-screw vessel fitted with Curtis turbines, with geared cruising turbines and designed for a speed of 29½ knots at about 1,090 tons trial displacement with the main engines developing about 17,000 shaft horsepower. The vessel was built under contract by the Fore River Shipbuilding Co. of Quincy, Mass. The contract was signed September 22, 1913, the price being $861,000.00, and the time of construction twenty-four months.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	310–00
on L.W.L., feet and inches	310–00
overall, feet and inches	315–03
Breadth, extreme, over guards, feet and inches	30–06½
on L.W.L., feet and inches	29–10
Draught to L.W.L., feet and inches	9–05
Mean trial displacement, tons	1,090
Ratio, length to beam	10.39
beam to length	.0962
Ton per inch immersion at L.W.L.	13.8

BATTERY.

There are four 4-inch rapid-fire guns in the battery, one on the forecastle, two on the main deck forward, port and starboard at break of forecastle, and the other on the main deck center line abaft the deck house.

The torpedo equipment consists of four 6.8-m by 21-inch

twin torpedo tubes mounted on the main deck, two port and two starboard.

There is a four-stage Ingersoll-Rand air compressor for charging the torpedoes. It is located in the engine-room and has a capacity of 20 cubic feet of air at 2,500 pounds per square inch per hour. It is steam driven, the steam cylinder being 8 inches diameter and the compressor cylinders 9¼, 4¾, 3⅛ and 1¼ inches diameter, respectively, for the four stages, with a common stroke of 5 inches. There is one accumulator for each tube with a capacity of 8 cubic feet.

MACHINERY.

The propelling machinery consists of Curtis turbines arranged on two shafts as shown on Plate I. On each shaft there is a 36-inch pitch diameter, 5-stage geared cruising turbine connected to the main shaft by means of a Metten clutch, and a 63-inch pitch diameter, 42-stage ahead and a 9-stage astern turbine in a common casing.

The main turbines are designed to develop about 8,500 S.H.P. each when making about 580 revolutions per minute. The cruising turbines are designed to develop about 800 S.H.P. each at 2,352 revolutions per minute and are geared down in the ratio of 7.95 to 1. The cruising units may be used at all speeds ahead below about 20 knots and are disconnected for higher speeds ahead and while reversing.

The main ahead turbine contains forty-two stages, the first three being wheel stages and the other thirty-nine being single-bucket drum stages. Plate II shows the turbine assembled with casing, and Plate III, with the upper half of casing removed. The first stage has four moving rows of blades and the second and third stages three moving rows each. The third-stage wheel is forged integral with the forward drum, the other wheels being of the usual built-up construction.

There are two drums on the ahead turbine, the forward drum consists of stages 4–9, inclusive, and the after drum of stages 10–42, inclusive.

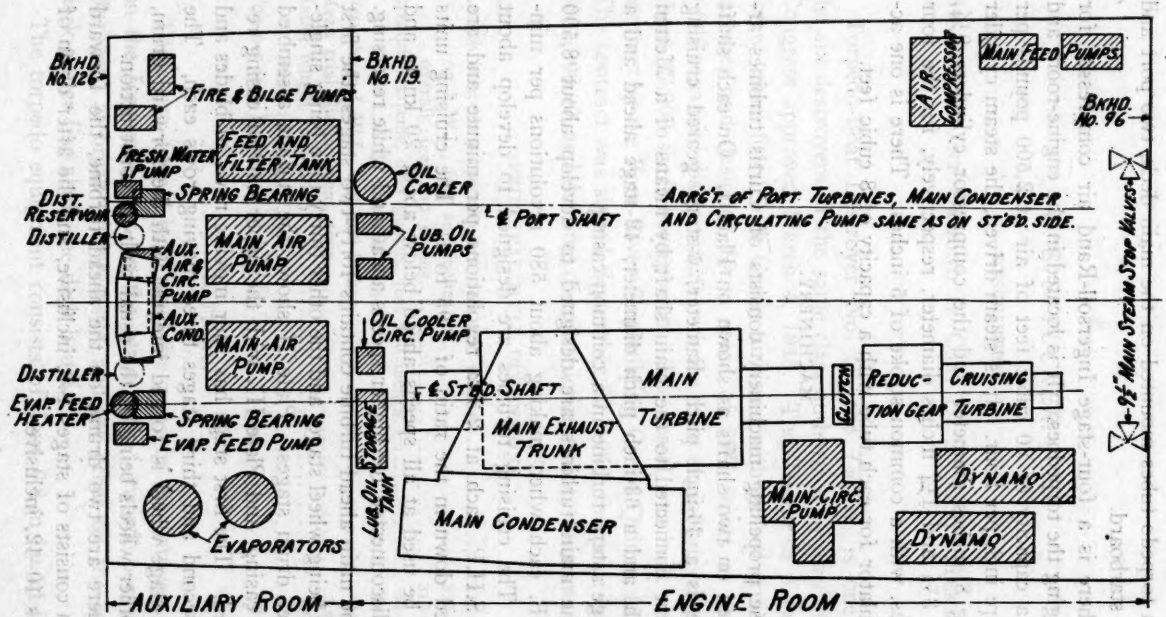


PLATE I.—U. S. S. "TUCKER."—ARRANGEMENT OF ENGINE AND AUXILIARY ROOMS.

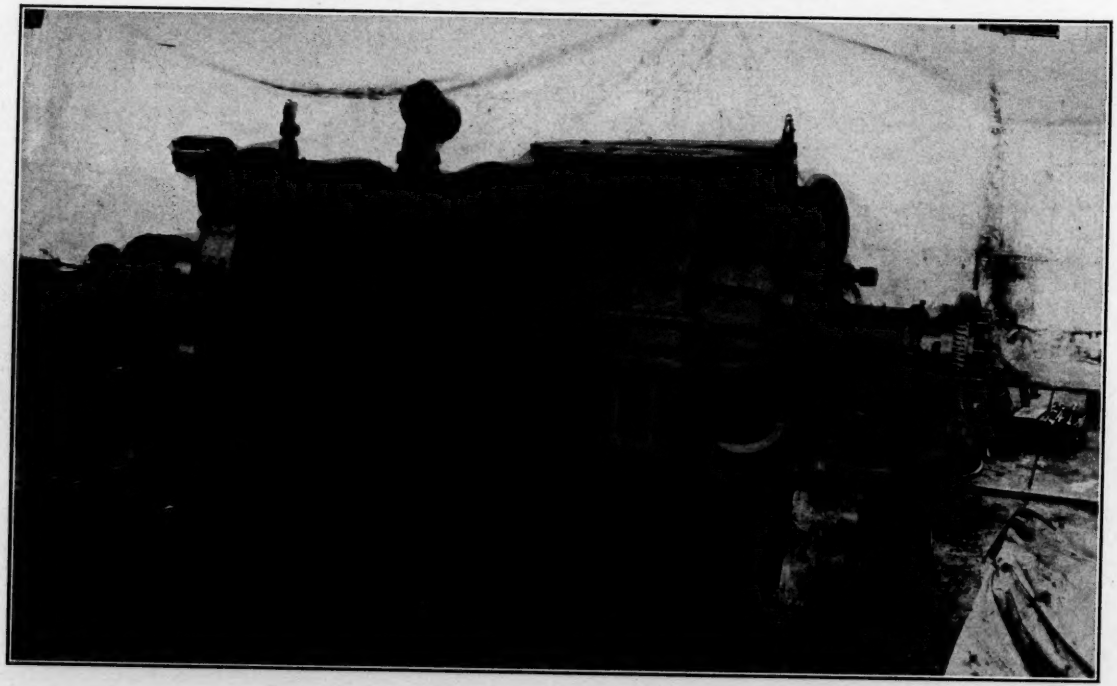


PLATE II.—U. S. S. "TUCKER"—MAIN TURBINE ASSEMBLED.

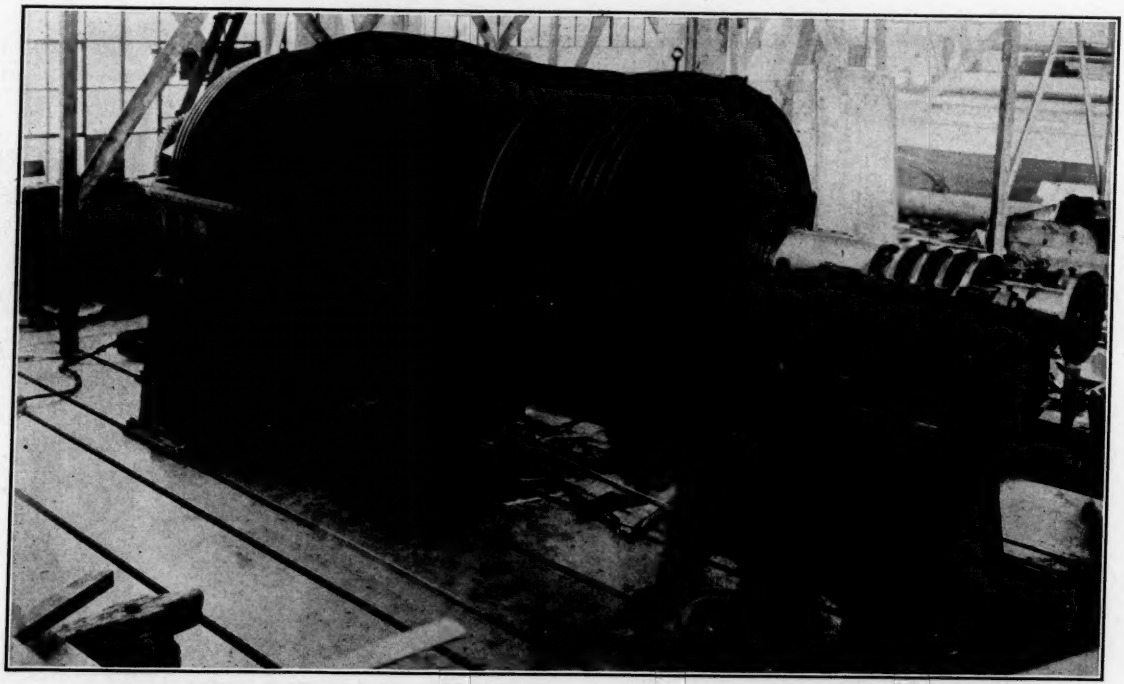


PLATE III.—U. S. S. "TUCKER."—MAIN TURBINE WITH UPPER HALF OF CASING REMOVED.

The reverse turbine consists of one stage with four moving rows of blades and a drum with eight single-bucket drum stages, and is capable of developing about 50 per cent. of the ahead power.

There are split, cast-steel diaphragms between the 1st and 2d, and between the 2d and 3d stages.

The nozzles for the 1st stage are of composition, those for the 2d and 3d stages of composition with monel-metal division plates. All other nozzles, blades and shrouding are rolled bronze.

All blades and the nozzles for the single-row stages are secured by means of dovetailed roots.

The casings are of close-grained cast iron, and are divided into six parts by a horizontal plane at the axis, and two athwartship planes.

On each shaft there is a 36-inch Curtis turbine, connected through gearing and a hydraulic clutch to the main shaft. It consists of five wheel stages, the first three having two moving rows of blades and the other two, one moving row of blades each.

The number of nozzles per stage, expansion ratio of nozzles and throat area in square inches is as follows:

CRUISING TURBINE.

Stage.	No.	E. R.	Throat area, square inches.	Height, inches.
1	8	1.182	1.576	.50
2	16	....	2.633	.50
3	40	....	6.804	.50
4	72	....	18.2412	.75
5	72	....	22.2912	.75
		Main Ahead.		
1	20	1.10	11.417	1.125
2	64	....	22.68	1.125
3	106	....	37.40	1.125
		Astern.		
1	14	2.00	9.45	1.50

The blades of the 1st stage ahead and astern are one inch in width; those for the 2d and 3d stages ahead are 7/8-inch wide, and for the 28th to 42d stages, inclusive, are 3/4 inch wide. All other blades are 6/10 inch wide. The nozzles for the single-row stages are the same width as the blades.

The effective length of the blades and the pitch diameter of the various stages is as follows:

Row.	Length, ins.	P. D., ins.
	1st Stage ahead.	
1	$1\frac{7}{16}$	
2	$1\frac{1}{8}$	
3	$2\frac{1}{4}$	
4	$2\frac{1}{8}$	63
5	3	
6	$3\frac{1}{8}$	
7	$3\frac{1}{4}$	
	2d and 3d stages ahead.	
1	$1\frac{7}{16}$	
2	$1\frac{13}{16}$	
3	$2\frac{9}{16}$	63
4	$2\frac{1}{8}$	
5	$2\frac{13}{16}$	
	1st stage astern.	
1	$1\frac{7}{8}$	
2	$2\frac{7}{32}$	
3	$2\frac{1}{32}$	
4	$2\frac{29}{32}$	63
5	$3\frac{9}{32}$	
6	$3\frac{5}{8}$	
7	4	

Stages.	Length, ins.	P. D., ins.
	Ahead turbine.	
4–6	$2\frac{3}{16}$	63
7–9	$2\frac{9}{16}$	$63\frac{1}{8}$
10–12	$3\frac{1}{8}$	$62\frac{5}{16}$
13–15	4	$62\frac{13}{16}$
16–18	$4\frac{1}{2}$	$63\frac{7}{16}$
19–21	$5\frac{1}{8}$	$63\frac{15}{16}$
22–24	$5\frac{5}{8}$	$64\frac{9}{16}$
25–27	$6\frac{1}{8}$	$65\frac{3}{16}$
28–30	$7\frac{1}{4}$	$66\frac{7}{16}$
31–33	$8\frac{1}{8}$	$67\frac{9}{16}$
34–36	$9\frac{1}{8}$	$66\frac{13}{16}$
37–39	$9\frac{5}{8}$	$67\frac{3}{16}$
40–42	$10\frac{1}{2}$	$67\frac{11}{16}$
	Astern turbine.	
2–3	$2\frac{7}{8}$	$60\frac{1}{4}$
4–5	$3\frac{13}{16}$	$61\frac{7}{16}$
6–7	$4\frac{1}{4}$	$62\frac{1}{8}$
8–9	$5\frac{1}{8}$	$63\frac{1}{4}$

SHAFTING AND BEARINGS.

Cruising-turbine rotor shaft, length, feet and inches	6–04
diameter, maximum, inches	$07\frac{1}{8}$
Main rotor and thrust shaft, length, feet and inches	22–$00\frac{1}{4}$
diameter, maximum, inches	$16\frac{9}{16}$
hole, inches	09
Forward line shaft, length, feet and inches	12–$01\frac{3}{16}$
diameter, inches	10
hole, inches	07
After line shaft, length, feet and inches	25–$02\frac{1}{4}$
diameter, inches	10
hole, inches	07

Stern-tube and propeller shaft, length, feet and inches	45–02½
diameter, inches	10
hole, inches	07
Bearings, main, number, each turbine	2
diameter, inches	13
length, inches	18
type	Babbitted.
Cruising, number, each turbine	2
diameter, inches	05
length, inches	07
type	Babbitted.
Spring, number	2
diameter, inches	10⅛
length, inches	14
type	Babbitted.
Stern-tube, forward, diameter, inches	10⅞
length, inches	26⅛
type	Lignum vitae.
aft, diameter, inches	10⅞
length, inches	43⅝
type	Lignum vitae.
Strut, forward, diameter, inches	10⅞
length, inches	31⅛
type	Lignum vitae.
after, diameter, inches	10⅞
length, inches	40⅝
type	Lignum vitae.

The main thrust bearing is at the forward end of the turbine shaft, and consists of four horseshoes of the regular type, babbitted on both faces and cored for the circulation of cooling water.

The turbine is so designed that the steam thrust balances the propeller thrust at a speed of about 27 knots. At higher speeds the steam thrust is the greater, and at lower speeds the propeller thrust is the greater. The total effective surface of the horseshoes is 219.0 square inches on forward side and 292.0 square inches on after side.

Number of collars	4
Thickness, inches	1¼
Space between collars, inches	03
Outside diameter, inches	14 3/16
Inside diameter, inches	08¾

The shaft horsepower developed by the engines is measured by a Gary-Cummings torsionmeter, one for each line of shafting.

The torsionmeter constants were found by calibration of the actual shafts with the meters mounted on them. The constant for the starboard shaft is 5.65 and for the port shaft 5.731.

PROPELLERS.

There is one three-bladed outboard-turning propeller on each shaft. The propellers are of manganese-bronze, cast solid, and the blades are true screw and machined to pitch and polished. The following are the actual propeller measurements:

	Starboard.	Port.
Diameter, feet and inches	7–09	7–09
Pitch, feet and inches	6–08.97	6–09.03
Ratio, pitch to diameter	.871	.871
Projected area, square feet	28.2	28.2
Helicoidal area, square feet	31.3	31.3
Disc area, square feet	47.17	47.17
Ratio, projected to disc	.5979	.5979
helicoidal to disc	.6636	.6636
projected to helicoidal	.901	.901
Lower tip of blade below keel, inches	05	05
Immersion of upper tip of blade at load draught, ins.	35¼	35¼

MAIN CONDENSERS.

There are two main condensers, pear shaped, of the curved-tube type, the tubes being rolled into the tube sheets. They are built up in the usual manner, the principal dimensions being as follows:

Length between tube sheets, feet and inches	14–00
of tubes as fitted, feet and inches	14–09
Thickness of tube sheets, inches	01¼
Tubes, number, each condenser	2,532
diameter, outside, inch	00⅝
thickness, B.W.G.	18
Cooling surface, square feet	5,800
Main exhaust nozzle	10 feet 04 inches by 22 inches.

Diameter, auxiliary exhaust nozzle, inches..... 07
dynamo exhaust nozzle (starboard only), inches..... 08
air-pump suction, inches..... 11
circulating-water inlet and outlet, inches..... 19½
drain connection, inches..... 03

TABLE I.—PUMPS AND CONNECTIONS, U. S. S. "TUCKER."

No.	Pumps.	Size (Ins.)	Type.	Suction Pipes from—	Ins.	Discharge Pipes to—	Ins.	Location.
2	Main Air.	12 x (2) 28 x 18.	Twin, Vertical, Bucket, Single-Acting, "Blake."	Condensers.	11	Feed Tank.	8	In Auxiliary Room.
				Reserve Feed Tanks.	1½			
				Condenser for Sealing Water.	1½			
2	Main Circulating.	33 Dia. Runner, 5½ x 3½ x 6 Engine.	Vertical, Double Engine, Centrifugal Pumps.	Sea.	14½	Main Condenser.	14½	In Engine Room.
				Bilge.	7			
2	Main Feed.	15 x 10 x 16	Vertical, Piston, Double-Acting, Single, "Blake."	Feed Suction Pipe and Res. Feed Tanks.	5½	Main Feed Disch.	4	In Engine Room.
				Cross-Conn. Pipe.	5½			
2	Aux. Feed.	Do.	Do.	Feed Suction Pipe and Res. Feed Tanks.	5½	Aux. Feed.	4	One in each Fire Room.
				Hose Connection.	1½	Hose Connection.	1½	
2	Fire and Bilge.	7 x 7 x 12	Do.	Sea.	4	Fire Main.	2½	In Auxiliary Room.
				Drainage.	4	Overboard.	3½	
				Hose Connection.	1½	Distiller, Flushing System thru Fire Main.	1½	
						Hose Connection.	1½	
2	Fire and Bilge.	Do.	Do	Sea.	3½	Fire Main.	2½	In #1 Fire Room.
				Drainage.	4	Overboard	3½	
				Hose Connection.	1½	Hose Connection.	1½	
				Independent Bilge.	4			
1	Aux. Air and Circ.	6 x 8 x 8 x 7	Combined, "Blake."	Aux. Condenser.	4	Feed Tanks.	3	In Auxiliary Room.
				Sea.	4	Aux. Condenser.	4	
1	Evaporator Feed.	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, "Blake."	Sea.	2	Evaporators.	1½	Do.
				Distiller Circ. Water Disch.	1½			
1	Distiller Fresh Water.	3½ x 4 x 4	Do.	Distiller Reservoir Tank.	1½	Res. Feed Tanks.	1½	Do.
						Ship's Tanks.	1½	
						Cofferdam.	1½	
						Main Feed Tank.	1½	
2	Lubricating Oil.	4½ x 6 x 6	Do.	Lubricating Oil Tank.	3	Bearings.	2½	In Engine Room.
						Oil Settling Tank.	1	
4	Fuel-Oil Service.	6 x 3½ x 8	Vertical, Piston, Double-Acting, Duplex, "Blake."	Oil Storage Tanks.	2½	Burners.	2	Two in each Fire Room.
				Booster-Pump Discharge.	2½			
2	Fuel-Oil Booster.	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, Light Service, "Blake."	Storage Tanks.	3	Service-Pump Suction.	2½	One in each Fire Room.
				Deck-Hose Conn.	2½	Storage Tanks.	2½	
						Deck Hose Conn.	2½	
2	Turbine Drain.	3½ x 4 x 4	Horizontal, Piston, Double-Acting, Single, "Blake."	Turbines.	1½	Filter Tank.	1½	One under each Main Turbine.
1	Oil to Clutch.	4½ x 2½ x 4	Vertical, Piston, Double-Acting, Duplex, "Blake."	Tank.	1	Clutch.	¼	In Engine Room.

MAIN AIR PUMPS.

Each main condenser is provided with a Blake vertical, twin-plex, single-acting beam air pump, located in the auxiliary room, with a steam cylinder 14 inches in diameter and wet and dry-air cylinders 28 inches each, with a common stroke of 18 inches. The wet suction is 9 inches in diameter and the dry 7 inches, both taking off the common 11-inch suction from the condenser. From the wet-air pump there is an 8-inch discharge to the feed and filter tank.

MAIN CIRCULATING PUMPS.

There is one centrifugal circulating pump for each main condenser, driven by a Sturtevant Type VD–6 steam turbine. The propeller is 3-bladed and has a diameter of 16 inches and a pitch of 12¾ inches.

AUXILIARY CONDENSER.

There is one auxiliary condenser, located in the auxiliary room. It is pear-shaped and of the curved-tube type, the tubes being expanded in the tube sheets.

There are 380 ⅝-inch tubes with a total cooling surface of 300 square feet. The length of the tubes as fitted varies from 5 feet to 5 feet 3⅛ inches between the tube sheets. Cooling water is supplied by an independent circulating pump. The pump has an impeller 18 inches in diameter, 4-inch suction and discharge, and is driven by a 3-inch by 3-inch single engine at 330 r.p.m.

There is a Blake featherweight air pump for the auxiliary condenser 4½ by 8 by 6.

The auxiliary exhaust connection to the condenser is 6 inches in diameter and the air-pump suction from the condenser 3½ inches.

FEED AND FILTER TANK.

This is located in the auxiliary room and has a total capacity of 818 gallons, of which 235 gallons belong to the filter chambers and 583 gallons to the feed tank proper. The tank has a 11-inch connection for main air-pump discharge, a 3-inch connection for the auxiliary air-pump discharge, a 8-inch feed-pump suction connection and a 4-inch overflow pipe. There is also a connection from the distiller fresh-water pump and 2-inch vapor pipes leading from each filter chamber.

FORCED–LUBRICATION SYSTEM.

The main turbine and thrust bearings, cruising-turbine bearings and gears are fitted with a complete system of forced lubrication. The system comprises the following:

2 225-gallon storage tanks.
1 200-gallon storage tank.
1 205-gallon oil-settling tank.
2 70-gallon drain tanks.
2 23-inch diameter Reilly multicoil oil coolers.
2 4½ by 6 by 6 Blake simplex, vertical oil pumps.

Oil is taken from the port drain tank by either of the oil pumps and discharged to the various bearings through the coolers. From the bearings the oil drains by gravity into a 6-inch pipe which runs from the forward bearing of the cruising turbine, aft through the bilge near the center line, to the starboard oil-drain tank, the two drain tanks being cross-connected by a 4-inch pipe.

Cooling water for the cooler is supplied by either the main circulating pumps or from the fire main.

Oil to the clutch is supplied by an independent 4½ by 2¾ by 4-inch duplex pump.

Each oil cooler has a cooling surface of 73.8 square feet.

BOILERS.

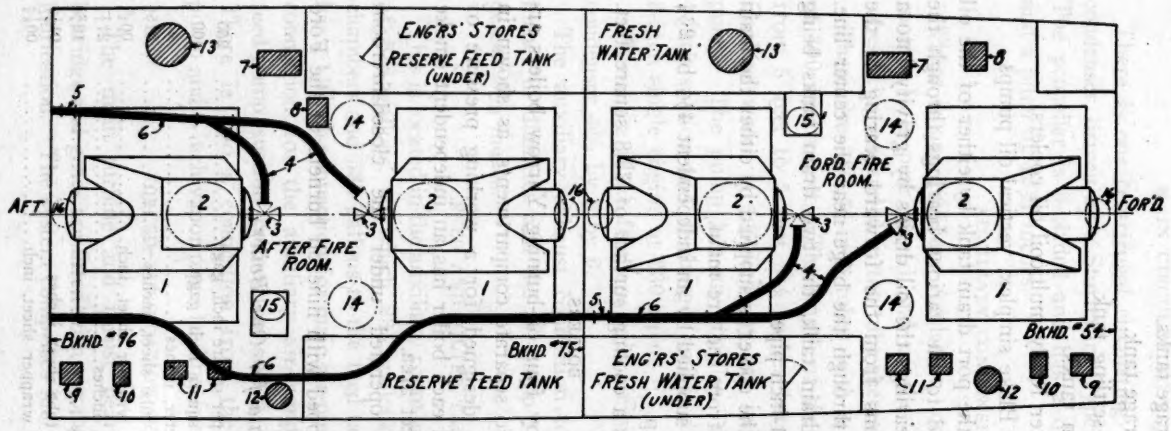
Steam is supplied by four oil-burning Yarrow boilers arranged in pairs in two separate compartments, as shown in Plate IV. They are designed for a working pressure of 265 pounds gage, and each boiler has an independent smoke pipe of 16.8 square feet area.

The firerooms are operated under the closed-fireroom forced-draft system.

Each boiler is equipped with nine oil burners of the Fore River type.

Data for One Boiler.

Pressure, working, pounds per square inch, gage	265
test, pounds per square inch, gage	400
Drum, steam, inside diameter, inches	43
thickness of tube sheet, inches	$0\frac{15}{16}$
wrapper sheet, inch	$0\frac{9}{16}$
water, inside width, inches	$27\frac{11}{16}$
depth, inches	$19\frac{13}{16}$
thickness of tube sheet, inches	$0\frac{1}{4}$
wrapper sheet, inch	$0\frac{1}{2}$



ITEM	DESCRIPTION	ITEM	DESCRIPTION
1	BOILER	9	FUEL OIL HAND PRESSURE PUMP
2	SMOKE PIPE	10	" " BOOSTER PUMP
3	7" MAIN STEAM STOP VALVE	11	" " SERVICE "
4	7½" O.D. MAIN STEAM PIPE	12	" " HEATER
5	MAIN STEAM EXPANSION JOINT	13	FEED WATER HEATER
6	10⅜" O.D. MAIN STEAM PIPE	14	TURBINE DRIVEN BLOWER
7	AUX. FEED PUMP	15	AIR LOCK
8	FIRE AND BILGE PUMP	16	ESCAPE HATCH

PLATE IV.—U. S. S. "TUCKER."—ARRANGEMENT OF FIREROOMS.

Number of 1-inch tubes	2,656
1¼-inch tubes	218
Heating surface, square feet	5,735
Furnace volume, cubic feet	630
Diameter of main stop valve, inches	07
feed, stop and check valves, one to each lower drum, inches	02
auxiliary feed, stop and check valve, inches	03
Diameter of surface-blow valve, inches	01¼
bottom-blow valves (two), inches	01½
safety valves (two duplex), inches	04

The auxiliary feed enters the steam drum and the main feed enters the water drums, where it is baffled so that it rises through the two outer rows of tubes.

All tubes are straight except the two rows nearest the furnace, which are curved slightly to provide for expansion. These latter are 1¼ inches external diameter, all others being 1 inch.

FUEL-OIL SYSTEM.

This system consists of two light-service booster pumps, one in each fireroom; four duplex service pumps, two in each fireroom; one oil heater in each fireroom, and the oil-storage tanks, valves and piping as required.

The booster pumps draw oil from the storage tanks and discharge to the suction of the service pumps. The latter draw either from the storage tanks direct or from the discharge line from the booster pumps and discharge to the burner lines.

There are nine burners per boiler, each supplied by a ⅜-inch branch from the burner line.

The whole system is completely fitted with strainers, stops and automatic stop-valves.

FUEL-OIL HEATERS.

There is a Reilly, vertical, multicoil fuel-oil heater in each fireroom. The heaters are 15 inches outside diameter, with four coils. Heating surface of coils, 23.64 square feet.

FORCED-DRAFT BLOWERS.

There are two 37-inch single-inlet Sturtevant cone fans in each fireroom, each driven by a 24-inch Curtis turbine direct connected to the fan. The turbine has one stage with three moving rows of blades.

FEED-WATER HEATERS.

There is one Reilly vertical, multicoil feed-water heater in each fireroom. The heaters are 36 inches in diameter and each contains 29 coils, with a heating surface of 178.35 square feet.

MAIN STEAM PIPING.

The main steam connection on each boiler is 7 inches in diameter, the forward boiler steam lines combine into one 9½-inch line which runs aft on the starboard side. The two after boilers combine into one 9½-inch line which runs aft on the port side, the two main lines being cross-connected in the engine room with a 6½-inch cross-connection. There are expansion joints in each line at the engine-room bulkhead and a stop valve at the engine-room bulkhead. Just inboard of the bulkhead stops there are 9½-inch astern and ahead turbine throttles and a 4-inch throttle for the cruising turbines.

AUXILIARY STEAM PIPING.

There is a 3½-inch stop valve on each boiler for supplying the auxiliary steam line. In each fireroom connections are taken off these lines for the various auxiliaries in the fireroom, the two lines in each fireroom then combine into a common 3½-inch line. In the after fireroom these lines are cross-connected and a 4-inch line runs aft on each side of the engine room to the auxiliary room, where they are cross-connected, forming a loop. From this line branches lead to the various auxiliaries.

AUXILIARY—EXHAUST PIPING.

The auxiliary-exhaust piping runs throughout the machinery spaces, varying in diameter from 2½ to 8 inches. There are branches to each auxiliary and the following connections:

4½ inches to each feed-water heater.
6½ inches to the 1st and 10th stage of each main turbine, the latter being 8½ inches in diameter at the turbine.
6 inches to the auxiliary condenser.
7 inches to each main condenser.
7½ inches to atmosphere.

MAIN AND AUXILIARY FEED SYSTEMS.

There is an 8-inch suction main running forward from the feed and filter tank with 5½-inch connection to the air-pump channel ways and a 5½-inch connection to each main feed pump in the engine room. The line runs forward to the after fireroom, where it divides into two 5½-inch branches, one to each auxiliary feed pump in the two firerooms.

The discharge from each feed pump is 4 inches in diameter, the discharges from the two main feed pumps combining into one 6-inch line which runs forward to the firerooms, with a 4-inch branch to each feed-water heater or by-pass, thence to the lower drums of the boilers.

Each auxiliary feed pump discharges direct through the auxiliary feed line to the upper drums of the boilers or cross-connects to the main line, thence through the heater or by-pass to the lower drums.

EVAPORATING AND DISTILLING PLANT.

There are two evaporators in the auxiliary room and two distillers in the hatch over the auxiliary room.

Data for One Evaporator.

Type	Reilly, vertical, multicoil
Diameter, inside, inches	40
Height overall, feet and inches	6–09

Number of coils	14
Diameter of coils, inches	04⅝
tubes, inch	01
Thickness of tubes, inch	00.065
Heating surface, square feet	86.10
Diameter of steam nozzle, inches	02
vapor nozzle, inches	03½
feed valve, inch	01
blow valve, inches	02

Data for One Distiller.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	24
Height overall, feet and inches	5–00
Number of coils	8
Diameter of coils, inches	04⅝
tubes, inch	01
Thickness of tubes, inch	00.065
Cooling surface, square feet	49.20
Diameter of vapor inlet, inches	02½
drain connection, inch	00¾
circulating-water connections, inches	02½

EVAPORATOR FEED-WATER HEATER.

There is one Reilly, vertical, multicoil feed-water heater located in the auxiliary room. It is 13 inches inside diameter with three coils with a heating surface of 18.45 square feet.

ELECTRIC PLANT.

The dynamos are located in the forward starboard corner of the engine room. The installation consists of two horizontal, compound wound, direct-current, 25-k.w., General Electric generators, each driven by a Curtis steam turbine. Each generator will deliver at normal load 200 ampères of current at 125 volts, when running 3,600 revolutions per minute.

U. S. S. *BUSHNELL*, SUBMARINE TENDER No. 2.

DESCRIPTION.

By Lieutenant G. E. Davis, U. S. N., Member.

Contract for the construction of the *Bushnell* was awarded to the Seattle Construction & Dry Dock Company, Seattle, Washington, the contract price being $935,695. Delivery was to be made within 21 months from date of contract, June 30, 1913.

PRINCIPAL HULL DIMENSIONS.

Length on L.W.L., feet and inches	300–00
over all, feet and inches	350–10⅞
Beam L.W.L., feet and inches	45–06
extreme, feet and inches	45–08
Displacement corresponding, tons	3,575
Tons per inch immersion at L.W.L.	23
Area immersed, midship section, square feet	665
Load water plane, square feet	9,690
Co-efficient of fineness block	0.611
Midship section	0.976
L.W.L.	0.710

GENERAL DESCRIPTION.

The hull of this vessel is constructed in accordance with the highest rating under Lloyds Rules, the machinery being built in accordance with the rules of the American Bureau of Shipping for vessels of the highest rating, and with the rules of the U. S. Steamboat Inspection Service for ocean-going vessels. The inspection was carried on by naval inspectors appointed by the Department, as well as by the inspectors of the various societies which later classified this vessel.

MAIN DECK.

The main deck is continuous from stem to stern. On it are located a forward deck house containing the commanding of-

ficer's and division commander's quarters and offices; a midship deck house containing paymaster's and executive officer's offices, galleys, bakery, bread room, butcher shop, potato-peeler room, foundry, blacksmith shop, coppersmith shop, and a coal bunker with coal supply for the galley and bakery ranges and ovens; the after deck house, which contains the sick bay, dispensary and a well-equipped operating room. Above the deck house forward is located the chart house and the radio room, while above these still is the navigating bridge. Two 24-inch distant mechanically-controlled searchlights are located on platforms on the outboard ends of the bridge about eight feet above the same. A range-finder platform is located above the bridge and over the radio room on a slightly higher level than the searchlight platforms. A 500-gallon fresh-water gravity tank is located on the midship deck house just forward of the smokepipe.

On the after deck house are located vegetable lockers, gasoline stowage, hand-steering station, and two searchlight platforms similar to those forward, on which are mounted two 24-inch distant mechanically-controlled searchlights.

Forward of the after deck house on the main deck are provided stowage racks for stowing gasoline drums for motorboat fuel.

SECOND DECK.

The second deck is continuous from stem to stern. Forward are located the paint rooms, crew's heads, washrooms, a well-equipped laundry, ship's store, general mess pantry and crew's quarters. Amidship are the machine shop and torpedo-testing room, the equipment of which will be described later, while aft are the wardroom officers' quarters.

PLATFORM DECK.

This deck extends from the stem to the fireroom and from the engine room to the stern. Forward are located the paint room, sail room, chain lockers, C. P. O. mess room, C. P. O.

berthing space, G. S. K. clothing and small-stores rooms, refrigerator rooms and crew's space, while aft are the torpedo armory, warrant officers' quarters, wardroom staterooms and officers' storerooms.

HOLD.

In the forward hold are the trimming tanks, electrolyte storeroom, chain locker and fuel tanks, and G. S. K. storerooms, in one of the latter of which are portable fresh-water and battery-water tanks. The boiler room, fuel and settling tanks, engine room and engineer's storerooms occupy the entire space in the midship hold, while aft are located the magazines, war-head rooms, ordnance storerooms, fresh-water tanks and lubricating-oil tank and trimming tank.

DOUBLE-BOTTTOM AND OIL TANKS.

The practically complete double bottom is used for fuel-oil stowage, reserve feed water, fresh water, and lubricating oil. In addition to the double bottoms fuel oil is stowed in three deep tanks forward of the fireroom and in two deep settling tanks outboard of the fireroom. Cofferdams are provided between oil tanks and water tanks and machinery spaces.

BATTERY.

The battery consists of four 5-inch, 50-caliber rapid-fire guns, two 3-pounder saluting guns, and one 18-inch twin-deck torpedo tube, located in the torpedo-testing room on the second deck.

FRESH WATER.

Potable fresh water is stowed in tanks forward in the G. S. K. storeroom to the extent of fifteen tons, and aft in the double bottoms to the extent of 62.4 tons. Reserve-feed water tanks in the double bottom beneath the fireroom have a capacity of 48.6 tons, while the feed and filter tanks has a normal capacity of 3.36 tons. A 500-gallon gravity tank is located on the up-

per deck forward of the smokestack as previously mentioned, while in a forward G. S. K. storeroom is provided a lead-lined battery water tank of 1,000 gallons capacity.

DRAINAGE SYSTEM.

Fire Main.—A 6-inch fire main is supplied by the fire and bilge pump as an emergency supply from the distiller circulating pump.

All hose connections at fire plugs are 2½-inch.

Hose connections are also provided on the suction and discharge sides of fire and bilge pump.

VENTILATION AND HEATING.

Combined mechanical heating and ventilation is provided for all spaces below the main deck, except the bathrooms, toilet heads and washrooms, which are provided with radiators. Five forced-ventilation fans with steam coil-heating boxes are provided for this purpose. By-passes are provided for the air around these boxes for the supply of certain compartments, such as magazines which it is not desired to heat. In addition to the five fans above referred to, two forced-ventilation fans are provided for engine-room supply and one exhaust fan for crew's heads and wash rooms. All spaces within the deck houses, chart house and radio room are provided with radiators and natural ventilation.

BOATS.

The following boats are provided and stowed on the upper-deck level on boat skids between the forward and the midship deck houses, and the midship and after deck houses:

3 35-foot motor boats (torpedo chasers).
2 33-foot motor sailing launches.
2 21-foot motor dories.
2 14-foot punts.

All the above-mentioned boats are handled by means of cargo booms on the fore and main masts.

In addition to those listed above, two 28-foot whale boats are provided and stowed outboard of the midship deck house. These boats are handled by davits.

ANCHOR WINDLASS, TOWING ENGINE AND SUBMARINE LIFTING GEAR.

There is fitted forward on the main deck a windlass of special design, of the horizontal spindle type, with two gypsies and three wildcats, for handling the 7,000-pound anchors. An arrangement of sheaves and blocks in the bow is provided, making it possible to lift a weight of 30 tons, which permits the lifting of the stern of a submarine out of the water for an inspection of propellers, etc. A similar gear is fitted to the towing engine aft on the main deck, making it possible to lift a dead weight of ten tons. To facilitate the handling of weights as noted above the vessel is provided with a cableship type of bow and a stern of large overhang.

COMPLEMENT.

1 Division commander (submarines).
1 Commanding officer (tender).
17 Wardroom officers (submarines and tender).
3 Warrant officers (tender).
26 Chief petty officers (submarines and tender).
275 Enlisted men (submarines and tender).

MACHINERY INSTALLATION.

(See Plate I for arrangement of machinery spaces.)

The main propelling machinery consists of a high-pressure and low-pressure Parsons marine steam turbine, developing collectively about 2,500 shaft horsepower running at 2,000 r.p.m., connected to the single line shaft through the medium of a common mechanical reduction gear of gear ratio about 16.421 to 1. The after end of each turbine shaft is connected

1- H.P. TURBINE
2- L.P. "
3- REDUCTION GEAR
4- THRUST BEARING
5- MAIN CONDENSER
6- AUX. "
7- MAIN CIRC. PUMP
8- AUX. " "
9- MAIN AIR "
10- AUX. " "
11- MAIN FEED "
12- AUX. " "
13- GREASE EXTRACTOR
14- FEED WATER HEATER
15- FEED TANK
16- LUB. OIL PUMPS
17- " " COOLER
18- ICE MACHINE
19- EVAPORATORS
20- EVAP. FEED PUMP
21- SANITARY PUMP
22- DISTILLER F.W. PUMP
23- DIST. CIRC. WATER "
24- ELECT. FRESH WATER "
25- FIRE AND BILGE PUMP
26- BILGE PUMP
27- TURBO BLOWER
28- FUEL OIL BOOSTER PUMP
29- " " SERVICE "
30- " " FILM HEATER
31- BOILERS.
32- TURBO GENERATORS, 50 K.W.
33- MAIN SWITCH BOARD
34- I.C. MOT. GENERATOR, 20 V.
35- I.C. SWITCH BOARD
36- I.C. STORAGE BATTERY 20 V.
37- BATTERY CHARGING GEN. 300 K.W.
38- OIL COOLER CIRC. WATER PUMP.
39- AIR COMPRESSORS

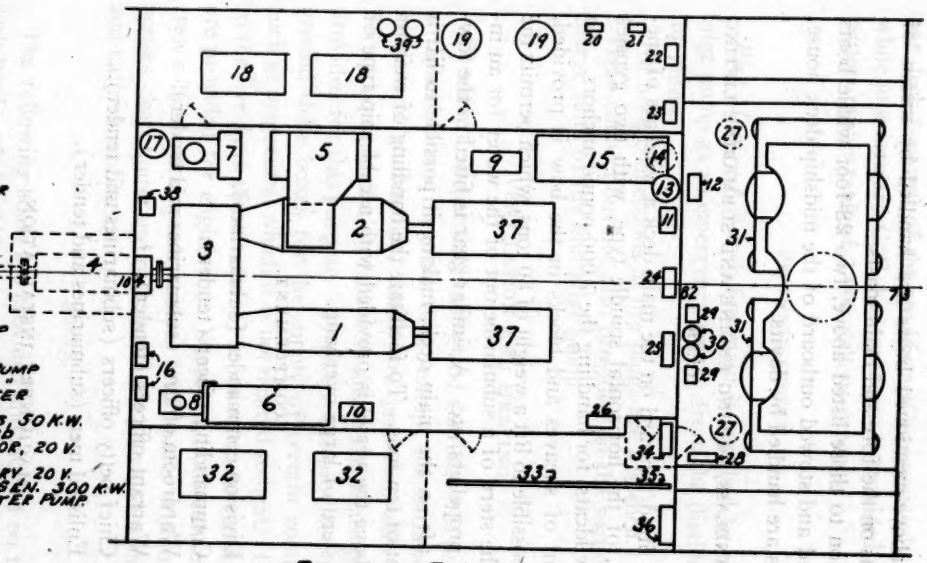


PLATE I

U.S.S. BUSHNELL - SUBMARINE TENDER NO. 2.

ARRANGEMENT OF MACHINERY

to its separate reduction-gear pinion by means of a claw-clutch coupling. An astern turbine, capable of developing 50 per cent. of full-head power, is incorporated in the after end of the low-pressure turbine.

BOILER EQUIPMENT.

Boiler equipment consists of two Yarrow water-tube (express) boilers, W. P. 220 pounds per square inch, fitted for burning fuel oil, and working under forced-draft closed-stokehold system.

MAIN TURBINES.

The main turbines are of the usual Parsons marine steam type, consisting of a special grade cast-iron cylinder divided horizontally along the center line, the lower half supporting the turbine rotor bearings at each end. The high-pressure turbine has six stages, the low-pressure turbine six stages, and the astern turbine five stages.

Turbine Data.

Stage.	Diameter of cylinder. H.P.	L.P.	Astern.	Length of cylinder (this diameter). H.P.	L.P.	Astern.
	Ins.	*Ins.*	*Ins.*	*Ins.*	*Ins.*	*Ins.*
1	15½	30½	19	13⅞	5⅝	7½
2	16½	32	20	19½	5½	8¼
3	17	34½	22	19½	5⅝	8¼
4	23½	38	25	9 15/16	7	7⅞
5	24½	38	25	9 11/16	7	7½
6	25½	38	...	9 11/16	7½	...
	Rows of blading.			**Length of blades.**		
1	15	4	6	0.97	2.345	.72
2	15	4	6	1.282	3.215	1.215
3	15	4	6	1.72	4.46	2.21
4	7	4	5	1.345	6.33	3.705
5	7	4	5	1.845	6.33	3.705
6	7	4	...	2.47	6.33	...
	Thickness of calking sections.					
1	9/32	9/32	9/32			
2	9/32	9/32	9/32			
3	9/32	9/32	9/32			
4	9/32	13/32	9/32			
5	9/32	13/32	9/32			
6	9/32	13/32	...			

Reduction Gear.—The reduction gear consists of a heavy cast-iron casing in which is mounted a large herring-bone gear which meshes with small pinions on each side. The bearings, which are of the spherically-seated type in the case of the pinions, and of the straight type in case of the gear-wheel shaft, are supported in bearing brackets which form a part of the lower half of the casing.

Gear Wheel.—The gear wheel consists of a cast-steel spider divided along the center and held together on the shaft by means of two heavy forged-steel shrink rings and properly keyed to the shaft. On the outside of this cast-steel spider are shrunk two forged-steel rings which are keyed and pinned to same. The teeth of the gear are cut in these forged-steel rings, and consist of 312 teeth, 3 diametral pitch, tooth twist 45 degress, total face length parallel to axis of shaft, 32 inches, pitch circle diameter 103.820. The pinions are a special alloy steel forged solid with the shaft and have 19 teeth each, of 3 diametral pitch, tooth twist 45 degrees, total face length on line parallel to shaft axis, 32 inches, pitch circle diameter 6.350. Each pinion is supported by three bearings, the forward and after bearings being 5½ inches x 5½ inches, the center bearing 5½ inches x 10 inches. The gear-wheel shaft bearings consist of 12½ inches x 12½ inches straight-type fitted at the fore and after ends of the shaft.

Lubrication.—All bearings are lubricated by the forced-lubrication system, the gear-wheel shaft bearings being provided with indicators, the pinion bearings with test cocks. The point of contact between the gear wheel and the pinions is kept well lubricated by means of 16 oil sprayers or nozzles which throw a goodly stream of oil directly on to the point of contact when the gears are running ahead.

TURBINE THRUST BEARINGS.

The turbines, being of the balanced type, need but very small thrust bearings, which are fitted at the forward end of the turbine casing and are of the usual Parsons type.

Main Thrust Bearing.—The main thrust bearing is located directly aft of the reduction gear to the gear-wheel shaft to which the thrust shaft is coupled.

Thrust Bearing Data.

Thrust shaft bearings:	
Length of each, inches	11½
Collars on shaft:	
Number	9
Thickness, inches	1⅞
Space between, inches	4
Outside diameter, inches	20
Inside diameter, inches	12
Effective bearing surface, square inches	1,809
Thrust shoes:	
Number	9
Effective thrust surface, square inches	1,269

The thrust shoes are of the usual horseshoe type, babbitted on both faces and cored for circulation of cooling water.

TORSION METER.

In the section of line shafting directly aft of the thrust shaft is fitted a Gary-Cummings torsion meter.

SHAFTING AND BEARINGS.

High-pressure rotor shaft	15′ 3⅝″ long, 5½″ diameter.
Low-pressure rotor shaft	16′ ¾″ long 6″ diameter.
Thrust shaft	9′ long, 12″ diameter.
Line shaft	3 sections, 14′ long, 11½″ diameter.
Line-shaft section, fitted with torsion meter, 14′ long, 11½″ diameter, 4″ hole.	
Propeller shaft	26′ ⅜″ long, 12½″ diameter.
Rotor-shaft bearing diameters:	
H.P. forward, inches	10½
H.P. aft, inches	10½
L.P. forward, inches	12½
L.P. aft, inches	12½
Line shaft, or spring bearings:	
Four	12 inches long.
Stern-tube bearings:	
Forward bearing, length, inches	14¾
Aft bearing, length, inches	60

PROPELLER.

There is one four-bladed, right-hand, built up, adjustable, true-screw, manganese-bronze propeller, diameter, 12 feet 6 inches.

Pitch as set, mean, feet and inches	14–3
Pitch, adjustable from	13 feet to 15 feet 6 inches.
Ratio of diameter to mean pitch	.877
Area projected, square feet	41.66
Area helicoidal, square feet	49
Area disc, square feet	122.71
Height of lower tip of blade above bottom of keel, inches	12
Immersion of upper tip of blade at load draught, inches	18

MAIN CONDENSER.

There is one main condenser, circular shape, straight-tube type, tubes being secured in tube sheets by Navy standard type of screwed ferrule, packed with cotton-tape packing thoroughly saturated with paraffine wax. It is on the usual built-up steel-shell type, with cast-iron heads, the principal dimensions being as follows:

Length between tube sheets, feet and inches	8–00
of tubes fitted, feet and inches	8–02⅛
Thickness of tube sheets, inches	01
Number of tubes	2,570
Diameter of tubes, outside, inch	00⅝
Thickness of tubes, B.W.G., No	16
Cooling surface, square feet	3,365
Main exhaust nozzle area, square feet	18

Condenser is provided with one wet-suction 7 inches in diameter, and a dry-suction of the same diameter. Also with auxiliary exhaust nozzle, 7 inches in diameter, and the usual air cocks, relief valves, drains, manholes, hand holes, etc.

A 5-gallon boiler compound tank is situated directly above the main condenser and connected thereto by means of a 1-inch pipe, fitted with a valve.

Circulating-water inlet and outlet, diameter, 14 inches.

MAIN AIR PUMP.

One main air pump of the Blake, vertical, simplex, beam type is provided. It has a 9-inch diameter suction from the water seal or trap below the main condenser to which the wet suction, augmentor-condenser drain, and turbine drain, of a 7-inch, 6-inch, and 2-inch diameter, respectively, are led. The wet suction only is trapped, the dry suction and turbine drains entering the trap on the pump side of the water seal. A 2-inch non-return valve opening toward the seal is fitted in the turbine drain pipe to the water seal.

MAIN CIRCULATING PUMP.

There is one main circulating pump of the centrifugal type, 14-inch diameter, driven by a vertical single-cylinder reciprocating engine. All moving parts of this engine are completely enclosed and provided with forced lubrication from the main forced-lubrication system.

AUGMENTOR INSTALLATION.

A Parsons augmentor installation is provided, the augmentor having a 7-inch suction connected to the dry suction of the condenser, a ¾-inch steam supply pipe leading to a jet having a nozzle opening 7/32-inch diameter, and discharging into an augmentor condenser through a 6-inch opening. The augmentor condenser, in turn, drains to the water seal through a 6-inch pipe.

Augmentor Condenser.—The augmentor is of the usual single-flow type, consisting of a cast-iron shell of circular section, having straight tubes, the principal dimensions being as follows:

Length between tube sheets, feet and inches	3–00
of tubes as fitted, feet and inches	3–02⅛
Thickness of tube sheets, inch	1
Number of tubes	205
Thickness, B.W.G., No	16
Diameter outside, inch	00⅝
Cooling surface, square feet	101.5

Augmentor discharge to condenser, diameter, inches	06
Condenser drain to water seal, diameter, inches	06
Circulating water inlet and outlet, diameter, inches	03

Circulating water is supplied from the main circulating pump, and discharges via the main overboard discharge.

Auxiliary Condenser.—The auxiliary condenser is of the built-up steel-shell type, similar to the main condenser, the principal dimensions being as follows:

Length between tube sheets, feet and inches	8–05¼
of tubes fitted, feet and inches	8–07⅞
Thickness of tube sheets, inch	01
Number of tubes	792
Diameter of tubes, O.D., inch	005⅛
Thickness of tubes, mils	65
Cooling surface, square feet	1,100
Circulating water inlet and outlet, diameter, inches	07

It is provided with the following nozzles:

50-k.w. turbine generators, exhaust pipe, 14 inches diameter.

Auxiliary exhaust discharge to auxiliary condenser, 7 inches diameter.

Main H.P. turbine exhaust, 12 inches diameter.

A 5-gallon boiler compound tank, similar to that on the main condenser, is fitted to the auxiliary condenser.

Auxiliary Air Pump.—One auxiliary air pump of the Blake vertical, simplex, featherweight type is provided. It has a 5-inch diameter suction from the auxiliary condenser and a 4-inch discharge to the feed and filter tank.

Auxiliary Circulating Pump.—One auxiliary circulating pump of the centrifugal type, driven by a vertical single-cylinder reciprocating engine, supplies cooling water to the auxiliary condenser. It is provided with a 7-inch suction nozzle and a 7-inch discharge to the condenser.

FEED SYSTEM.

Three modes of supplying feed water to the boilers are provided, viz: main feed system, auxiliary feed system, and injector.

Main Feed System.—The main feed system is served by a main feed pump located in the forward part of the engine room. This pump is of the Blake, vertical, simplex, piston type. It is provided with a 4-inch suction manifold, whereby it can take a suction from the feed and filter tank or from either of the reserve feed-water tanks. It discharges to the main feed system by way of the 3-inch pipe through a Riley multi-screen feed-water filter and grease extractor, and a 25-inch Riley multicoil feed-water heater. A by-pass is provided around this latter. From the feed heater a 3-inch pipe is led to the fireroom, where four 1½-inch branches are taken therefrom which lead to the lower water pockets of each boiler, discharging into a perforated pipe running the entire length of the pocket. This pipe is encased in such a manner that practically all feed water is deflected to the two outer rows of tubes, *i. e.*, the two rows of tubes farthest from the furnaces, from whence it discharges to the steam drum. The feed pipes are provided with the usual stop valves and check valves at the boiler.

Feed Water Data.

Number of coils	19
Thickness, B.W.G., No.	16
Outside diameter, inch	00⅞
Heating surface, square feet	80
Length of coil, feet and inches	2–10
Number of turns	28
Outside diameter of coil, inches	04

Auxiliary Feed System and Injector.—The auxiliary feed system, served by the auxiliary feed pump, which is similar to the main feed pump, and by a 2-inch Pemberthy auto-positive injector, both located in the fireroom, takes its feed by way of a 4-inch suction from either of the reserve feed tanks or from the feed and filter tank, and discharges through a 3-inch main from which 2-inch branches are led to each boiler. The auxiliary feed enters the lower part of the steam drum, discharging into a 2-inch perforated internal feed

pipe running the entire length of the drum. The injector is provided with a 2-inch suction from the feed and filter tank, with a 2-inch discharge to boilers by way of the auxiliary feed system.

Boiler Blows.—Each boiler is provided with one 1½-inch surface blow, leading from the steam drum, and two 1½-inch bottom blows, one from the bottom of each water pocket. A connection is provided from the boiler blow pipe whereby the boilers may be pumped out by means of the auxiliary feed pump through a 3-inch suction, the auxiliary feed pump being provided with a 3-inch discharge overboard to facilitate this latter.

Make-up Feed Connections.—A make-up feed connection is provided from both reserve feed tanks to the main air-pump suction.

FORCED-LUBRICATION SYSTEM.

All bearings of main turbines, reduction gear, 300-k.w. generators and main circulating-pump engine, are provided with forced lubrication. This system is supplied by two pumps of the Blake, vertical, simplex, piston type. Cooling water for the oil cooler is supplied by a similar pump. All bearings drain to the lubricating-oil drain tank, a tank of about 300 gallons capacity built into the double bottom. Each bearing drain to this tank is provided with a sight glass, which makes it possible to readily observe the flow of oil from any bearing or group of bearings. The forced-lubrication pumps take their suction from this tank through a suction strainer discharging through the oil cooler to the various parts requiring lubrication.

The oil cooler is a 4-flow, straight-tube type, having a cooling surface of 252 square feet.

A reserve lubricating-oil tank is provided, having a capacity of 200 gallons.

A settling tank of 200 gallons capacity provided with a steam coil is installed for the purpose of extracting water, grit, etc., from the oil.

A suction is provided from oil tank in the after double bottom whereby the lubricating-oil pumps may be used to transfer oil from this storage tank to a submarine alongside, by way of hose connections on the main deck at frame 104, port and starboard. Deck connections are provided for filling all tanks. In addition to the lubricating-oil storage above mentioned, tanks are provided in the after engine-room storeroom for engine oil, cylinder oil and lamp oil; on the dynamo flat for dynamo oil, and on the auxiliary flat, port, for ice-machine oil. These tanks supply the necessary oil for hand lubrication. All tanks are provided with a common deck filling connection and with suitable drains and cocks so that the proper oil only may be distributed to the desired tank.

FUEL OIL SYSTEM.

Fuel oil is stored in seven double-bottom tanks forward, three deep tanks just forward of the fireroom, two deep settling tanks outboard of the fireroom, and six double-bottom tanks beneath the engine room and aft. Two 4-inch Navy standard fuel-oil filling connections are provided on each side of the vessel, those on the port side being at frames 75 and 99, and those on the starboard side at frames 75 and 103. These connections combine in a single 6-inch lead to the fuel-oil suction and filling manifold at frame 73 in the fireroom, and with a 6-inch transfer connection leading to the after manifold in the engine room at frame 96 starboard. All tanks are provided with ample vents and with sounding pipes, and all pipes from deep tanks have valves outside the tanks, operated from the second deck.

PUMPS AND CONNECTIONS.

Booster Pump.—A single booster pump of Blake, vertical, simplex, piston type is installed in the fireroom on the port side. The suction and discharge connections are so arranged that oil may be transferred from the forward tanks to the

after tanks or vice versa. The deep settling tanks may be filled from either the forward or the after tanks by way of the settling tank low suction to which a branch from the booster pump discharge is led. The booster pump is also provided with a discharge to filling connections on the ship's sides by way of a 3-inch meter. This connection is for the purpose of supplying oil for submarines alongside. A connection is also provided from the discharge side of this pump for filling the foundry fuel-oil furnace storage tank located on the main deck within the foundry space.

Fuel-Oil Service Pumps.—Two fuel-oil service pumps of the Blake, vertical, duplex, piston type are installed. Fuel oil may be taken by these pumps from either the settling-tank high suction, the settling-tank low suction, or the transfer line through a 2½-inch duplex Schutte & Koerting suction strainer. A suction connection is also provided whereby the suction from the settling tank low suction may be by-passed around the suction strainer. These pumps discharge through a 1½-inch meter, which is by-passed to two Schutte & Koerting spirally-corrugated film fuel-oil heaters, also by-passed, and through a 1½-inch Schutte & Koerting duplex discharge strainer to the burners. A discharge branch from these pumps is led overboard, so that water pumped from the settling tanks by way of the settling tank low suction may be discharged overboard. Thermometers are fitted at the inlet and outlet of the heater; relief valves are fitted to the discharge side of the pump, and a governor fitted in the steam connection to the pump. The steam connections to the fuel-oil service pumps and the fuel-oil booster pump are so arranged that they may be shut off from the second deck.

Condensed steam from the fuel-oil heaters is led through a bucket trap to an inspection tank, from whence it is drained to the feed and filter tank. This inspection tank is provided with a float valve so arranged that the water level is always kept well above the drain outlet, so that leaks in the oil heater will be indicated by the presence of a film of oil on top of the water in this inspection tank.

FORCED-DRAFT BLOWERS.

Two forced-draft blowers, consisting of a vertical type ZVM Terry turbine directly coupled to a 25-inch open Keith fan, are installed. Each blower is capable of delivering 12,000 cubic feet of air per minute against a head of 3 inches of water, when running at 1,600 r.p.m. These turbines are designed to operate at 200 pounds steam pressure and 10 pounds back pressure. Each turbine develops normally about 13 B.H.P.

MAIN STEAM PIPING.

A 4½-inch main steam pipe is led from a stop valve on each boiler to a common Y fitting at the after fireroom bulkhead, from whence a 6-inch pipe is led to the main turbine maneuvering manifold. From this manifold leads are taken as follows:

5½-inch to high-pressure turbine;
5-inch to low-pressure turbine;
4-inch to astern turbine.

Double-beat throttle valves are fitted in each of these lines at the maneuvering manifold, and in addition, governor valves of the piston type are fitted in the lines to the high-pressure ahead and low-pressure ahead turbines. These valves and their connecting mechanism are described later under electric plant. The low-pressure ahead and astern throttle valves are interlocked in such a way that it is impossible to admit steam to both turbines at the same time, which operation might strip the blades.

AUXILIARY STEAM PIPE.

An auxiliary steam pipe with 4-inch branches from each boiler is led in a complete circuit around the engine and firerooms with cutouts in suitable locations. All steam for auxiliary machinery is taken from this line, either directly or through reducing valves.

MAIN EXHAUST.

The high-pressure turbine is so connected that it may exhaust to the auxiliary condenser, to the main condenser through the low-pressure turbine casing, and to the steam chest of the low-pressure turbine. The low-pressure turbine exhaust trunk leads direct to the main condenser.

AUXILIARY EXHAUST.

The auxiliary exhaust pipe forms a complete circuit around the engine room and is provided with a branch in the fireroom. The exhaust of all auxiliaries, except the two 50-k.w. turbo generators, which exhaust directly to the auxiliary condensers, is taken care of by this line, which has outlets and discharges as follows:

Auxiliary condenser;
Main condenser;
L.P. turbine casing;
Feed water heater;
To the atmosphere, by way of the fireroom branch.

The connections at the condensers and to the turbine casings are provided with spring-loaded back-pressure valves. The other connections are fitted with ordinary stop valves.

EVAPORATOR AND DISTILLER PLANT.

An evaporating and distilling plant is installed, capable of making 7,500 gallons of water per day of 24 hours. The connections are so arranged that the plant may be worked either single or double effect. The steam generated in the shell of either evaporator may be directed either into the auxiliary exhaust line or to the distillers as desired. The distillers are located in the engine hatch, starboard side, forward. They drain to a small distiller-drain tank directly below the distiller fresh-water pump, which transfers the water to the ship's tanks, or this tank may be drained by

gravity to the feed and filter tank or to the reserve feed-water tanks.

Distiller and Evaporator Data.—Two distillers, 20-inch inside diameter, Riley multicoil, Type D, 23 feet 6 inches, of 1-inch O. D. copper tubing, .065 inch thick, made into coils of 25 turns each, 4⅝ inch O. D., 3 feet 4 inches long, 8 coils per distiller.

Two evaporators, Riley multicoil type, 36-inch, I. D., 23 feet 2 inches of 1-inch O. D. copper tubing, No. 16 B. W. G. thick, made into coils of 24 turns, each 4⅝ inches O. D., 3 feet 3 inches long, 12 coils per evaporator.

REFRIGERATOR PLANT.

A complete refrigerator and ice-machine plant is installed, consisting of two one-ton Allen dense-air refrigerating or ice machines, horizontal, steam-driven type, located on the port engine room flat aft, two ice-making boxes, six No. 40 cans each, located on the second deck, port side, and three cold-storage rooms, located on the platform deck forward.

COMPRESSED-AIR WORKSHOP SYSTEM.

A No. 100 compressed-air system is installed for workshop tools, blowing boiler tubes, boiler fuel-oil atomization for port use, foundry fuel-oil furnace atomization, fuel-oil air chamber charging, operation of submarine signal bell, etc. Outlets are provided in the boiler room, machine shop, blacksmith shop, coppersmith shop and foundry. This system is supplied by two Westinghouse, vertical, single-stage, single-cylinder, steam-driven air compressors of a capacity of 45 cubic feet of free air per minute, located on the port engine-room flat. Accumulators of ample size are fitted in the boiler room and in the machine shop. Compressors are provided with automatic constant-pressure governors. A branch from the high-pressure torpedo compressed-air system is connected to this system through a suitable reducing valve.

TORPEDO COMPRESSED-AIR SYSTEM.

A high-pressure torpedo compressed-air system is installed for the purpose of charging torpedoes and charging air banks of submarines alongside, and for salvaging purposes if necessary. A main is run fore and aft which is provided with branches leading to six outlets, three on each side of the vessel, forward, amidship and aft. Portable charging pipes are provided for connecting to air banks of submarines and for supplying air for salvaging purposes, as desired. This system is supplied by two electric-driven torpedo-air compressors of the Platt Iron Works manufacture, installed in the torpedo-testing room port and starboard. They are of the 4-stage type, of a capacity of 50 cubic feet per hour at 2,500 pounds per square inch. Air accumulators of the total capacity of approximately 25 cubic feet are installed in the torpedo-testing room.

MACHINE SHOP.

A well-equipped machine shop is provided, capable of taking care of the ordinary run of repair work required on submarines. The equipment consists of the following:

One hand wood trimmer.

One pattern maker's lathe, extension gap type, 18 inches by 24 inches, with gap closed, capable of taking 6 feet between centers with gap open, and swinging 34 inches.

One 32-inch band saw.

One power-driven hacksaw, capacity 7-inch by 8-inch solids.

One 16-inch sensitive drill press.

One 14-inch by 7-foot tool-room lathe.

One double emery grinder, wheels 2 inches by 12 inches.

One 14-inch by 5-foot tool-room lathe.

One 2½-inch radial drill.

One 16-inch by 32-inch extension gap lathe (engine), capable of taking 7 feet 2 inches between centers when gap is extended and swinging 32 inches through the gap.

One universal milling machine, No. 1 B. Hendey type,

longitudinal travel of table, 32 inches; transverse travel of table, 6 inches; vertical travel of table, 18 inches.

One 15-inch Potter & Johnson Universal tool-room shaper.

A tool room is installed amidships, provided with necessary tool racks, benches, etc. Work benches with vises, both wood and metal, are installed in convenient locations.

TORPEDO-TESTING ROOM.

An 8-inch back-geared precision lathe of the Rivett type is installed in the torpedo-testing room for work on torpedo parts.

FOUNDRY.

A well-equipped foundry is installed on the main deck, provided with a Mircs fuel-oil furnace, and having the usual molding benches, flasks, racks, and complete equipment of foundry tools.

BLACKSMITH SHOP.

The blacksmith shop is installed in the after part of midship deck house, port, and supplied with one No. 4 Beaudry Peerless power hammer, rated at 100 pounds, electric driven. One Buffalo down-draft electric-driven forge. It is equipped with the usual work benches, anvils, racks, tools, etc.

COPPERSMITH SHOP.

The coppersmith shop is installed on the main deck in the midship deck house, aft, on starboard side. It is equipped with the following tools:

One hand-operated forming machine with 3-foot rolls.

One hand-operated seaming machine to take work of 3 feet length.

It is supplied with the usual hand tools, work benches, metal racks, etc.

ELECTRIC PLANT.

The electrical system is supplied with current from two 50 k.w. generators, each driven by two-stage, condensing, horizontal Curtiss steam turbines, running at 3,300 r.p.m.,

and from two 300-k.w. generators, each driven by one of the ship's main propelling engines, which consists in the case of the starboard engine of a six-stage, high-pressure, Parsons marine steam turbine, while the port engine is a six-stage, low-pressure turbine of the same type. A five-stage astern turbine is incorporated in the after end of this low-pressure turbine. The 300-k.w. generators are fitted at the forward end of these turbines and direct connected to them through 4-inch claw clutch couplings.

MAIN TURBINE GENERATORS AND OPERATING GEAR.

Two governors are provided for each of the ahead turbines, one being connected to each turbine rotor shaft, the other to each 300-k.w. generator shaft. The first mentioned governor is intended for propeller duty only, and is so adjusted that when the speed of the turbine exceeds the normal by a predetermined amount, *i. e.*, when a speed of 2,250 r.p.m. is reached, the turbine governors, acting through a system of levers and cranks, cause a butterfly valve in the main 6-inch steam pipe to close. The generator governors are so adjusted that when on generator duty they perform the same function when the generators reach a speed of 1,600 r.p.m., or 100 r.p.m. above the normal speed. In addition these latter governors, acting through a system of levers attached to a relay type of sliding piston-governor valve, regulate the speed of the turbines to compensate for load variation on the generators by throttling the steam admitted.

The two 50-k.w. machines supply direct current at 125 volts to the ship's lighting fixtures, receptacles, switches and receptacles, signal lights, running and anchor lights, searchlights, radio outfit, ventilation sets, deck winches, workshop machinery, fresh-water pump, laundry machinery, and to the interior communication systems.

The two 300-k.w. generators, which are primarily intended as battery-charging generators for charging storage batteries of submarines alongside, supply direct current at a voltage

which may be varied from 125 volts normal, when used for lighting purposes and for power, to 300 volts normal, when used for storage-battery charging. These machines are the only sources of current for the two torpedo-air compressors, as these compressors pull too large a load for the small machines.

The two-wire system of distribution is used throughout, the various circuits being fed direct from the main switchboard, located on the dynamo platform starboard side, forward of the 50-k.w. generators, and from auxiliary distribution boards, located in certain compartments or sections of the vessel; these distribution boards in turn feeding from the main switchboard. The total output of the plant is controlled from the main switchboard located as noted above.

The current supply for the entire interior communicating system is controlled from the interior-communication switchboard, located just forward of the main switchboard. This switchboard supplies 125 volts direct current to the circuits requiring same, and also to the interior-communication motor generator, and to the storage batteries for the interior-communication system which operate on 20 volts direct current. The low voltage from the I. C. motor generators and batteries is also distributed from the interior-communication board.

Storerooms under the dynamo platform are provided for spare parts of electrical apparatus, lamps, instruments and other necessary stores.

A complete radio outfit, provided by the government and installed by the contractors, is fitted, with switch panel in the radio room and duplicate motor generators with automatic starting panel installed on the second deck forward.

A complete submarine-signal apparatus is provided and installed. The bell is of the electro-pneumatic type, air being supplied to it from the workshop compressed-air system via a suitable reducing valve.

LUBRICATING OILS.

THE EXAMINATION, LABORATORY AND PRACTICAL METHODS OF TESTING.

By Lieutenant (Junior Grade) J. L. Kauffman, U. S. Navy, Member.

This article will deal with the lubricating oils used in marine service, having the special requirements of the naval service particularly in view.

Up to about five years ago the Navy Department issued specifications covering each oil required; these specifications called for a certain flash point, specific gravity and viscosity, and usually enumerated several simple tests that would be made. This practice was similar to the practice of all railroads and industrial corporations, and one which is still followed by a few large firms. A few years after the establishment of the Engineering Experiment Station at Annapolis, Md., the study of lubricating oils was taken up and simple tests were made on oils purchased for naval use. The continued examination of oils developed a number of special tests which finally led to a change in the method of purchasing lubricating oils.

Instead of requiring all oils submitted by prospective bidders to meet definite specifications, each oil was required to pass satisfactorily the tests made at the Engineering Experiment Station before they were placed on the acceptable list. A memorandum was issued to all firms bidding on the Navy contracts setting forth in a general way the tests that would be made on the various oils. The old method had the disadvantage that an oil could be submitted which would meet each

item of the specifications and still would not lubricate properly; also the requirements of the old specifications regarding specific gravity and flash point were found to limit the number of competing firms without increasing the lubricating value of the oils submitted. The memorandum issued by the Bureau of Steam Engineering of the Navy Department has been changed several times, each time broadening the requirements in a few particulars and narrowing the requirements in others.

The important paragraphs of the recent memorandum are given below.

(*a*) "All oils not known to be satisfactory are subject to test at the Engineering Experiment Station, Annapolis, Md., and to a further test in service on board ship when considered necessary, before being considered acceptable by the Bureau of Steam Engineering.

(*b*) "The complete test of an oil consists of three parts, namely: chemical, physical and practical tests.

(*c*) "To successfully pass the chemical tests all oils should be neutral in reaction and should not show the presence of moisture, matter insoluble in petroleum ether (hard asphalt), matter insoluble in ether alcohol (soft asphalt), free sulphur, charring or wax-like constituents, naphthenic acids, sulphonated oils, soap, resin or tarry constituents, the presence of which indicates adulteration or lack of proper refining. Except in oil for engines without forced lubrication, no traces of fixed oils (animal or vegetable fats) should be found.

(*d*) "In lubricating oil for main engines without forced lubrication, approved fixed oils, such as rape seed, olive, tallow, lard and neatsfoot oil may be used. When the above fixed oils are used they will be well refined with alkalies, unadulterated, containing a minimum of free fatty acids, with no moisture or gumming constituents. Olive oil should not have a high specific gravity. If satisfactory emulsifying results can be obtained with straight mineral oils on engines without forced lubrication they may be submitted for service test.

(*e*) "The physical tests applied to each oil are as follows:

"1. { Specific gravity—pyknometer.
Baumé gravity.

"2. Flash and fire point (Cleveland open cup, Pensky Martin closed cup).

"The flash point should not be below 300 F., closed cup; and for steam cylinder oil not below 450 F.

"3. Cold point.—The cold point should not be above 32 F., except for cylinder oils. For ice-machine oils it should be as low as possible, at least low enough for the operating condition of minus 15 F. in a dense-air ice machine.

"4. Viscosity.—The Saybolt Universal Standard Viscosimeter is used. The viscosity of the oils must be sufficient for the purpose intended, and, except for ice-machine oils, must not be less than 140 seconds at 100 F. Viscosity is taken at 100, 130 and 210 F.

"5. Emulsion tests.—Emulsion tests are made on all straight mineral oils, except cylinder oils. Four emulsion runs are made, using 40 c.c. of oil in each case and

(*a*) 40 c.c. of distilled water;

(*b*) 40 c.c. of salt water;

(*c*) 40 c.c. of normal caustic soda solution;

(*d*) 40 c.c. of boiling distilled water.

The mixture is stirred with a paddle for five minutes at 1,500 revolutions per minute, the mixture being kept at a temperature of 130 F. during the stirring and while settling out. On oils used with forced lubrication or on ice machines the oil must completely settle out in less than 20 minutes.

"6. Friction tests are made on the Experiment Station friction machine to determine the relative friction of various oils under varying conditions of speed, bearing-cap pressure and temperature.

(*f*) "Practical tests of oils for forced lubrication are made on the Engineering Experiment Station's 75-kw. turbo-generator, making 2,400 revolutions per minute, with the temperature of the bearing varying from 180 to 210 F. Tests of ice-machine oils are made on the ½-ton Allen Dense-Air Ice

Machine. These tests cover a working period of at least 100 hours.

(*g*) "Upon completion of the tests at the Engineering Experiment Station a service test is usually recommended on all but cylinder oils. These tests are made on ships in commission, and cover a period of several months.

(*h*) "Oils should be submitted as follows:

"1. Three oils (light, medium and heavy) to be suitable for turbines, all forms of forced-feed or splash lubrication, dynamo engines, and all internal-combustion engines (including Diesel type). These three oils should cover the required range of viscosity.

"2. Marine-engine oil suitable for main engines without forced lubrication. This oil should form an emulsion when mixed with water, and may be either a compounded oil or a straight mineral oil. This oil must feed efficiently with wick feed, and not be washed off when mixed with water.

"3. An oil for use on ice machines in both the compressor and expander cylinders. This oil should not give off an objectionable odor when used with the dense-air ice machine.

"4. An oil for steam cylinder lubrication. This oil must be a straight mineral oil.

"5. Oil for steam cylinders using superheated steam. This oil will be used on shore stations and should have a high flash point."

The tests enumerated above will be described in a general way.

CHEMICAL TESTS.

Moisture.—Heat 3 to 4 c.c. in a test tube (the walls of which have been thoroughly wet with oil) in a bath of liquid paraffine up to 300 F. Oils containing water will form emulsions on the walls, also cause foaming and spluttering. A test is also made with a mixture of oil and eosin to determine faint traces of moisture by changes of color. The presence of moisture is particularly undesirable in transformer oils, but

there is danger of its forming objectionable emulsions in any straight mineral oil.

Sulphur.—Boil about 50 c.c. of oil with a piece of bright metallic sodium for half an hour; add water, heat and stir until sodium is dissolved; pour off water and test remainder with a fresh 1 per cent. solution of sodium nitro prusside. If the mixture turns violet color the oil contains sulphur. When sulphur is found an additional test for sulphonated oils is made.

Acids or Alkalis.—Heat for ½ hour with frequent stirring 25 c.c. of oil and 50 c.c. of neutral distilled water. Test a few c.c. of the mixture first with methyl-orange to determine the acids, and another portion with phenol-phthalein for the determination of alkalis. Acids and alkalis cause emulsions. Acids also cause corrosion of journals and other metal parts.

Matter insoluble in ether alcohol.—Shake 11 c.c. of oil and 14 c.c. of ether alcohol (8 parts ether and 6 parts alcohol). After standing 12 hours, note the precipitate, if any, at bottom of cylinder. The precipitate will be asphalt, and even a trace would make the oil undesirable as a lubricant. Asphalt would cause scoring of journals and clogging of oil lines.

Matter insoluble in high-grade gasoline.—Shake 3 c.c. of oil and about 300 c.c. of high-grade gasoline (86–88 Baumé gravity). After standing 12 hours, note precipitate, if any, in bottom of glass. The precipitate will be soft asphalt or carbon particles, and a slight trace would make the oil undesirable.

Tarry or Suspended Matter.—Same as above, except using 5 c.c. of oil and 95 c.c. of gasoline, and allowing it to stand for ½ hour; then examine deposit, if any, for dirt or tarry matter.

To Detect Fixed Oils.—Heat 10 c.c. of oil with a small piece of metallic sodium. If the mixture becomes gelatinized or a semi-solid it indicates the presence of fixed oils. If an equal volume of oil is heated alone to the same temperature the viscosity of the two samples can be compared; if the oil contains fixed oils (animal or vegetable oils) the sample with sodium will be much heavier than the sample heated alone. Straight mineral oils are used for all naval purposes afloat ex-

cept on marine engines *without* forced lubrication; and experiments are being made to determine whether or not a straight mineral oil can be used for the latter; an oil being required which will emulsify and not wash off when mixed with water.

Effect of Heat.—Heat 5 c.c. of oil in test tube over flame until vapors are evolved, and compare color of heated oil with color of unheated oils. If the heated oil turns black it shows the presence of undesirable carbon or hydrocarbons.

Additional tests are made, depending on the purposes for which an oil is to be used. If it is a compound marine-engine oil the name of the fixed oils and the percentage used is determined; the percentage of free acid in the fixed oil is also determined, as all fixed oils contain a certain percentage of free acid, but usually not sufficient to cause corrosion of the journals. A quantitative test for sulphur can also be made by burning the oil in a special lamp. A distillation test has been made to determine the percentage of light and heavy fractions contained in various mineral oils. This test is being continued, using samples of "used" oils.

PHYSICAL CONSTANTS.

The physical characteristics are determined as follows:

Color.—The color, although having no influence on the lubricating value, may be used to identify the sample. American oils fluoresce with a grass-green color, Russian oils have a blue sheen; oils containing distillation residues and unfiltered oils are brown to green-black in reflected light. Nearly all mineral machinery oils are distilled and filtered to some extent and are transparent in a test tube, the colors ranging from a yellowish white to a blood red. The color may be determined in a tinctometer by comparing with different colored glasses or lenses. These glasses are numbered and for machinery oil extend from a number 1 (white) to number 6 (red).

Odor.—The odor may be determined by heating in a test

tube or by rubbing on the hand, by which means fatty oils, coal tar, rosin oils, etc., may be detected.

Specific Gravity.—The specific gravity is obtained by the use of the "pyknometer," this term signifying any vessel in which an accurately measured volume of liquid can be weighed. The bottle is first filled with distilled water at a temperature of 60 F., and the weight of the water determined. The bottle is then filled with oil at a temperature of 60 F. and the weight of oil determined. The weight of oil divided by the weight of water gives the specific gravity at 60 F. The Baumé gravity is obtained by using the Baumé hydrometer, which is simply an ordinary hydrometer with a certain arbitrary scale. Baumé gravity may be converted into specific gravity by the following formula:

$$\text{Sp. gr.} = \frac{140}{130 + \text{Baumé}}.$$

Baumé gravity is largely used in commercial practice.

The specific gravity does not affect the lubricating value of an oil, but indicates to the experienced oil man the locality from which the crude oil is obtained. For instance, the specific gravities of the lubricating oils tested at the Experiment Station vary from 0.864 to 0.945. A Baumé gravity of 32 corresponds to a specific gravity of 0.864, and a Baumé gravity of 18.1 to a specific gravity of 0.945, so that an increase in specific gravity is a decrease in Baumé gravity. The paraffine base oils of Pennsylvania derivation have an *average* specific gravity of 0.875 with a corresponding Baumé gravity of 30. The asphaltic base oils from Texas and California have an *average* specific gravity of 0.930 with a corresponding Baumé gravity of 20.

Figure I (a) shows the pyknometer and (b) a Baumé hydrometer immersed in oil.

Flash Point.—The flash point is determined with both the Cleveland open cup (Fig. II, *a*) and the Pensky-Martin closed cup (Fig. II, *b*). The flash point of all oils is determined as a

measure of the volatility of the oils. The flash point of steam-cylinder oils is of primary importance, the required flash point depending on the temperature of the steam at the engine. With lubricating oils for bearings the flash point is important only in that it indicates the volatility of the oils, and the presence of kerosene or naphtha fractions, with the accompanying fire risks. In the case of very low flash-point lubricating oils it is desirable to run a special distillation or volatility test mentioned under chemical tests. The flash point determined with the open cup is higher than with the closed cup, as the inflammable gases on the surface of the oil are disturbed by the air currents in the open cup. These differences range from 5 degrees to 40 degrees, with the average at 20 degrees. The presence of very light ends (kerosene, naphtha, etc.) may increase this difference to 100 degrees.

Fire Point.—This is the temperature at which the oil burns, and is determined by raising the temperature about 3 degrees a minute, applying the flame for about a second. The fire or burning point is from 30 degrees to 65 degrees higher than the flash point with all lubricating oils, the light oils having a difference of about 40 degrees.

Cold Point.—Mineral oils become more viscous on cooling, and finally solidify. In lubricating oils refined from paraffine-base crudes, cooling first causes the paraffine particles to solidify, which gives the oil a cloudy appearance; with this class of oils this change is known as the cloud point.

The Committee on Lubricants of the American Society for Testing Materials uses the words "cold test" as a general term with sub-heads of "cloud test" and "pour test."

The method recommended by this committee is used at the Experiment Station, and in substance is as follows: Heat the oil to 150 F. and cool by air to 75 F. Take a bottle about 1¼-inch inside diameter and 4 to 5 inches high and pour in oil to a height of 1¼ inches from the bottom. Insert a cold-test thermometer (specially made, using colored alcohol, and with a long bulb) through a tight fitting cork. A special jacket

is used having an inside diameter about ½ inch larger than the bottle. Ice or any other cooling medium is packed around this jacket. When the oil is near the expected cloud point, at every 2 degrees drop in temperature remove the bottle and inspect the oil, being careful not to disturb the oil. When the lower half becomes opaque, read the thermometer; this reading is taken as the cloud point. The cold or pour test is simply a continuation of the cloud test, except that the temperature is noted every 5 degrees and the bottle tilted till the oil flows. When the oil becomes solid and will not flow, the previous 5 degree point is taken as the cold point of the oil. In the naval service a low cold point is required only on ice machines. All ice-machine oils should have a cold point of minus 15 F., which will permit a brine temperature of minus 20 F. Oils used on motor-boat engines should have zero cold test if the boats are used in cold climates. The lubricating oils refined from asphalt-base crudes, or Southern crudes, have a naturally cold point around zero degrees; these oils do not show cloudiness before solidifying. The oils from paraffine-base crudes, or Northern crudes, must be run through the filter presses to squeeze out the paraffine before a low cold test can be obtained.

Viscosity.—The viscosity of a lubricating oil is the most important constant to be determined. The viscosity of an oil is inversely proportional to its fluidity, and is a measure of its internal resistance or resistance to flow. The absolute viscosity is determined from the amounts flowing through capillary tubes, the results being given in C. G. S. units. The determination of the absolute viscosity is a very difficult operation requiring complex apparatus and a relatively long time. Several absolute viscosimeters have been invented; but to date none of them are considered practical enough for the routine testing of oil.

There are three well known commercial viscosimeters: the Engler, which is largely used in Europe; the Redwood, which is the English standard; and the *Salbolt Standard* Universal,

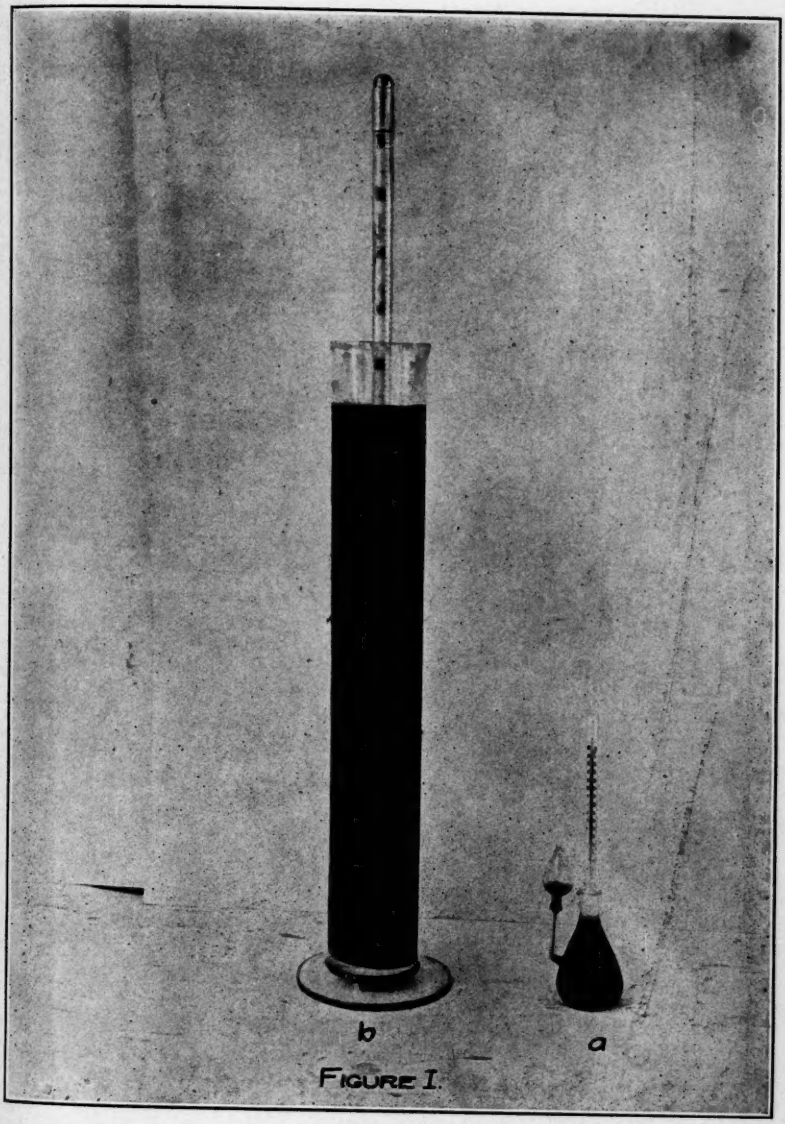


FIGURE I.

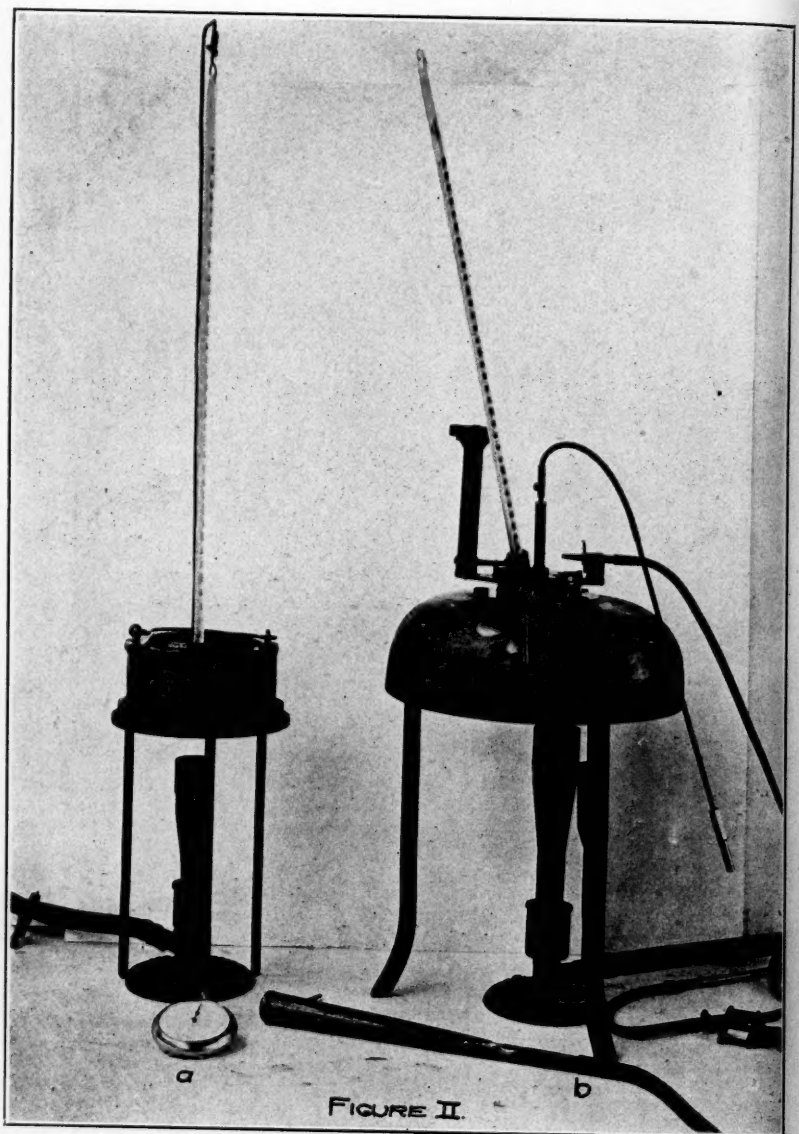


FIGURE II.

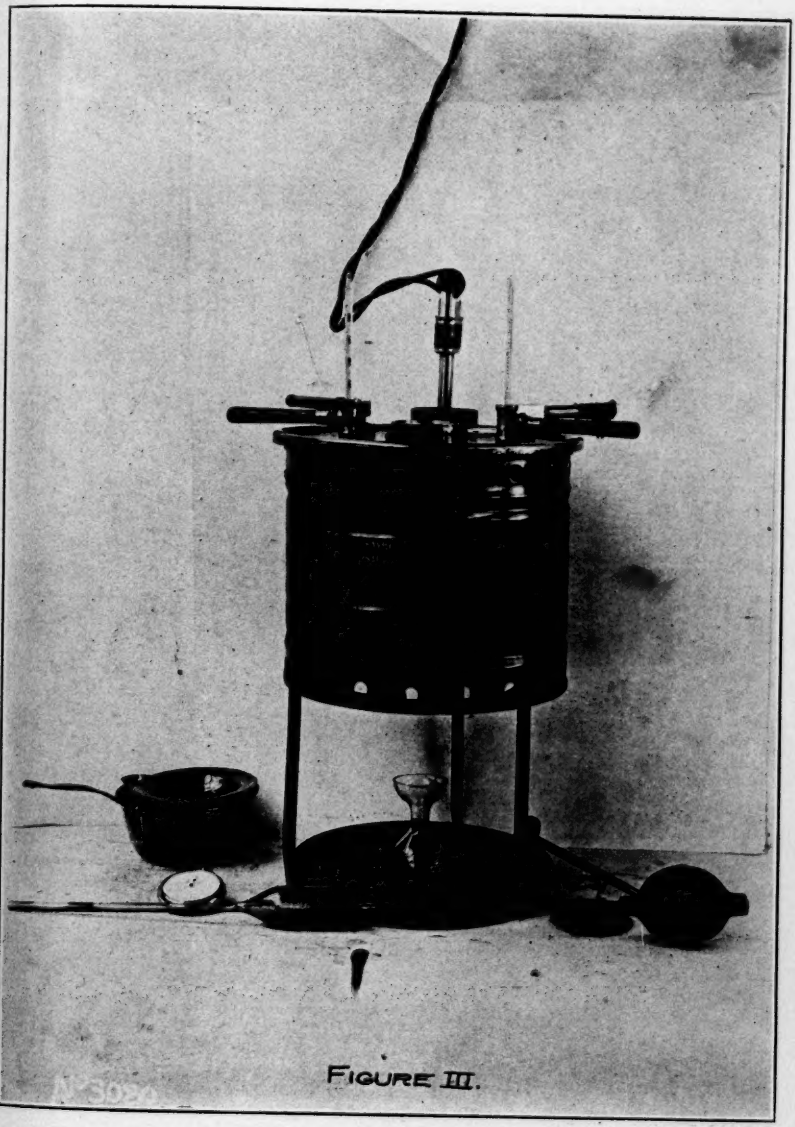


FIGURE III.

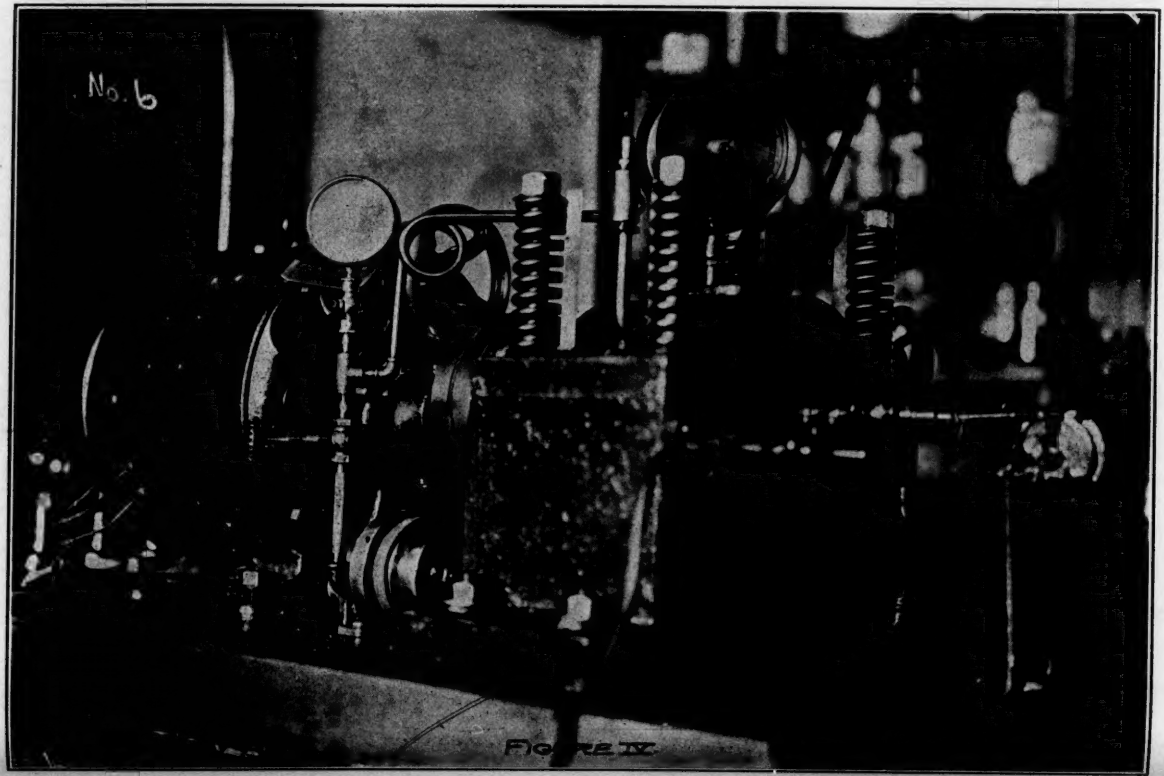


FIGURE IX

which is extensively used in this country. These three viscosimeters have been carefully compared at the Bureau of Standards by Dr. C. A. Waidner, and conversion tables have been prepared. The use of this type (the short capillary-tube type) has several disadvantages, the principal one being that the readings are simply comparative, and have no direct relation to the C. G. S. units of time, mass and length. The accepted theory advanced by Ubhelohde* is that the viscosity (absolute) is directly proportional to the internal friction of the lubricant, and that the viscosity is a direct indication of the friction which will be developed in a bearing. The use of the three commercial types (given above) is therefore allowable, as they will give the relative viscosity and corresponding friction of different oils. The readings of these viscosimeters may be converted to absolute viscosity units by using Ubhelohde's formulas, thus permitting the use of the commercial machines for scientific work, providing the proper corrections are made in regard to gravity and temperature.

Under normal conditions of bearing lubrication the lightest oil that will prevent seizing should be used to obtain a minimum frictional loss.

In Figure III is shown the Saybolt Universal Standard machine. At the Engineering Experiment Station four machines are used, on which very concordant results can be obtained.

Friction Machines.—Figure IV shows one of the friction machines designed and used at the Engineering Experiment Station. The machine consists of a hollow-steel journal, one foot in circumference and six inches in length, driven by a variable-speed motor. The bearing is of the usual white metal, and is lubricated by a forced-feed system, consisting of a small geared pump driven by a variable-speed motor; and a drain tank with cooling coils underneath the base of the machine. The load on the bearing is regulated by the calibrated springs shown in the photograph. An electrical heating unit is placed

*"The Theory of Lubrication, by L. Ubhelohde, "General Electric Review," November, 1915.

inside the journal so that the temperature of the journal may be varied.

Usually three series of runs are made. The first series is made with a constant speed of 600 revolutions per minute (in this machine also feet per minute), with a constant oil pressure of 15 pounds and with the bearing-cap pressures varying from 30 to 150 pounds per square inch. By measuring the watts input to the armature the coefficient of friction is obtained, knowing the load, speed, and armature position. The second series consists of runs with a constant bearing-cap pressure of 60 pounds per square inch, an oil pressure of 15 pounds, and speeds varying from 600 to 1,800 revolutions per minute. The third series consists of runs made with a constant bearing-cap pressure of 60 pounds per square inch, a constant speed of 600 revolutions per minute, an oil pressure of 15 pounds, and with varying bearing temperatures obtained by using the heating unit.

These tests are comparative only; the results being simply the friction obtained by using various oils on this *particular* bearing. Tests have been made which show that the friction is directly proportional to the viscosity of the oil at the temperature of the bearing.

Figure V shows the three friction curves for the various runs made on a light oil. This figure also contains viscosity and specific heat curves for this oil.

Emulsion Tests.—Emulsion tests are made on a machine designed by Lieutenant G. S. Bryan, U. S. Navy. An improved type is shown in Figure VI. The method of conducting the test is described in the memorandum quoted at the beginning of this article. The emulsion is made with distilled water, as there is always a possibility of fresh water getting in the system on board ship. Salt water is also used, as the oil coolers often leak, and in submarines the engine-cooling water seeps into the oil system. Normal caustic soda solution is taken, as there is a possibility of water containing boiler compound getting into the system. Boiling distilled water is used

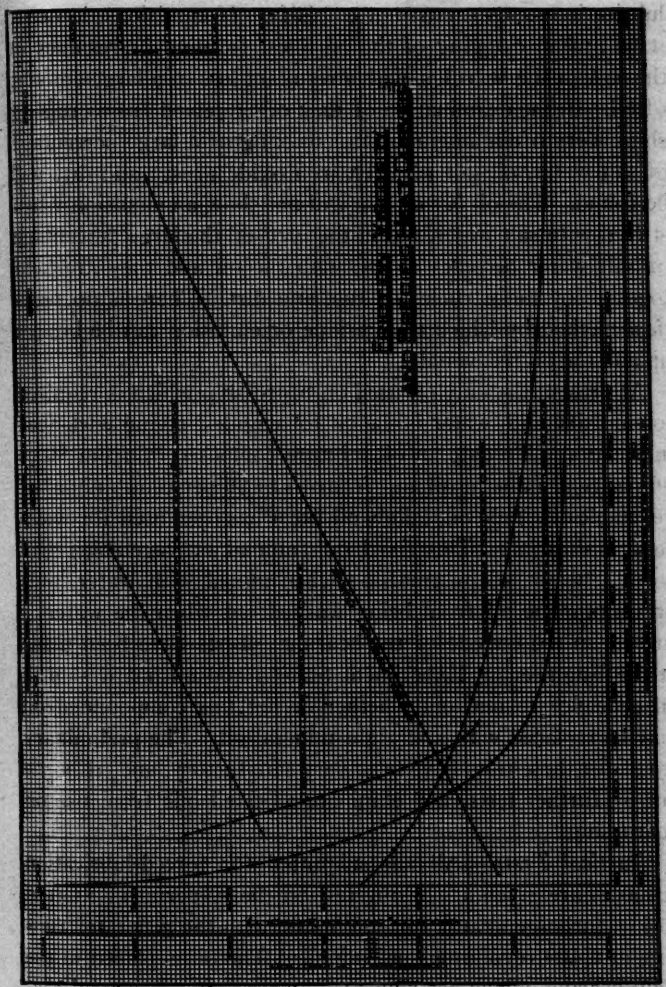


FIGURE V.

in case gland steam from turbines, or steam leaking by the piston-rod packing on reciprocating engines having forced lubrication, runs into the oil system.

These emulsion tests are considered of the greatest importance, as an oil to be used satisfactorily on any type of forced

lubrication system must not emulsify. If emulsions do occur it will mean clogging of the oil lines, forming of residues in the base of the bearings, with a resultant loss of a large amount of oil. In the examination of *used* oils received from cruising ships, very often an emulsion consisting of a large percentage of salt water, oil salt (NaCl), iron oxide (rust) and dirt will be found. The oil *before use* will not show an emulsion when stirred with salt water, but in the *used* oil the large percentage of salt water becomes mixed with the oil and they are held together by the particles of rust and dirt. There is no *practical* method by which this residue or emulsion can be broken up and the oil recovered.

At present the only oil used on board ship which is required to emulsify with water is the oil for engines having wick-feed lubrication. To date an oil mixture composed of from 80 to 90 per cent. mineral oil and from 10 to 20 per cent. animal or vegetable oil (rape oil, tallow oil, etc.) has been used for wick-feed lubrication. This fixed oil (animal or vegetable) will form a heavy, soapy emulsion when mixed with water, thus preventing the oil from being washed out when the water is turned on the outside of the bearing. Tests have been made at the Engineering Experiment Station and on board ship in an endeavor to find out whether or not this compounded oil may be replaced with a straight mineral oil. The preliminary results of these tests have been very satisfactory, and, if borne out by further tests, will eliminate all fixed oils from use on board ship.

Practical Tests.—All oils submitted for forced-lubrication purposes are given a working test on the Experiment Station 75-kw. turbo-generator for 100 hours, with the generator running at 2,400 revolutions per minute. The oil is placed in the main bearing, which has ring-feed lubrication. A careful record is made of the temperatures and of the amount of oil required to keep the reservoir full. After the test the bearing and journal are carefully examined; the oil is taken out from the various sections of the system, analyzed and tested. Natu-

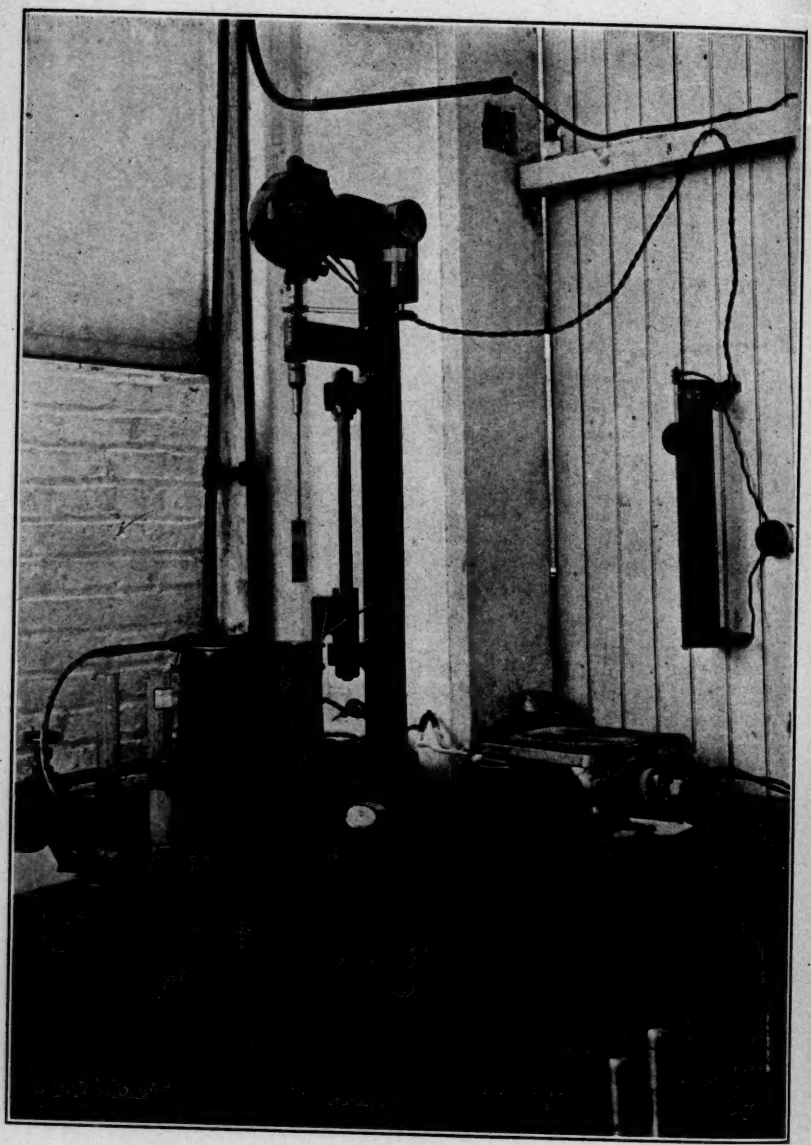


FIGURE VI.

rally a bearing of this type calls for a very light oil to give the best results. Nevertheless, all kinds of oils are tested on this bearing without resulting in any serious damage. From the examination after use in the turbine, the action of this oil in service can be predicted as far as its lubricating value is concerned.

Service Tests.—These tests are made on board ship. The oil to be tested is usually sent to a ship which has been using an oil which has proven satisfactory, and which has practically the same characteristics as the oil to be tested. This test extends over a period of several months, and should be prolonged for at least 500 hours of steaming. During the test the following data should be taken in order that the results may be properly interpreted:

Amount, in gallons, of oil put in system at beginning of test, and amount put in during test.

Maximum and average temperatures obtained on bearings during test.

Maximum and average temperature of oil entering the system.

Average pressure carried on lubricating-oil pumps.

Volume of oil passing through system at average cruising speed, in gallons per hour.

Amount of oil boiled out, and the loss due to this boiling out; also, temperature at which oil is boiled out.

Condition of bearings, tanks, and oil-feed lines after test.

Number of hours of steaming while oil was under test.

If, as is usually done, the oil to be tested is placed in the lubricating system of one engine room and the oil regularly used in the system of the other engine room the above data should be taken for both oils. After the test a 5-gallon sample of the new oil before use in the system, and a 5-gallon sample of the oil *after use* in the system, are forwarded to the Experiment Station for examination and test. This is done to insure that the oil furnished for service test is the same as the oil originally supplied to the station for preliminary test; also, for the purpose of studying the *used* oils.

Special Tests.—Due to the relatively small amount of information obtainable on the action of lubricants in service, it is necessary to continually develop new tests in order that the various new types of machinery may be properly lubricated.

At present the study of oils after several hundred hours' use in service is being made. Particular attention is being paid to the loss of oil by vaporization or volatilization, and the resulting characteristics of the remaining oil in the system.

One test on the "Fractional Distillation of Lubricating Oils" was described by one of the Experiment Station chemists, Mr. J. G. O'Neill, in the May, 1915, issue of the JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS. Further tests along this line are being made in an endeavor to find out the percentage loss after use in service of the various kinds of lubricants.

Tests are being made to determine the cause of emulsions, and methods of preventing this very undesirable condition.

Tests for carbon deposits in internal-combustion engines have been made with various oils, but the results are not sufficiently conclusive for publication.

A number of compounds have been tested which are supposed, when mixed with gasoline, to give increased power and speed, reduced carbon deposits and, at the same time, give efficient cylinder lubrication. The tests on these substances to date have not been wholly satisfactory.

Determination of the specific heat of various oils have been made in connection with oil-cooler tests. Also a test to determine the rate of flow through pipes will be made very shortly; the results to be used in the design of oil coolers and oil-feed piping.

At the present time it is a comparatively easy matter to obtain satisfactory oils for bearing lubrication by the employment of the tests above enumerated. From the large amount of data accumulated at the Experiment Station, whether or not an oil will prove satisfactory can be closely predicted by a consideration of the results obtained on the preliminary tests.

New mechanical features requiring new standards of lubrication are always coming up, which necessitate new tests. The lubrication of the *high-speed* reduction gears and the efficient cooling of Diesel engine pistons are two of the problems now requiring solution, which will undoubtedly be solved by making careful chemical, physical and practical tests of the various lubricants to be used in connection with a study of the method of applying the lubricant.

The tests described in this article have to a considerable extent been directly responsible for the present efficient lubrication of naval machinery, and a continued study will surely result in more efficient lubrication and in a great reduction in the amount of oil used per horsepower developed.

THE LJUNGSTRÖM SYSTEM OF ELECTRIC DRIVE.

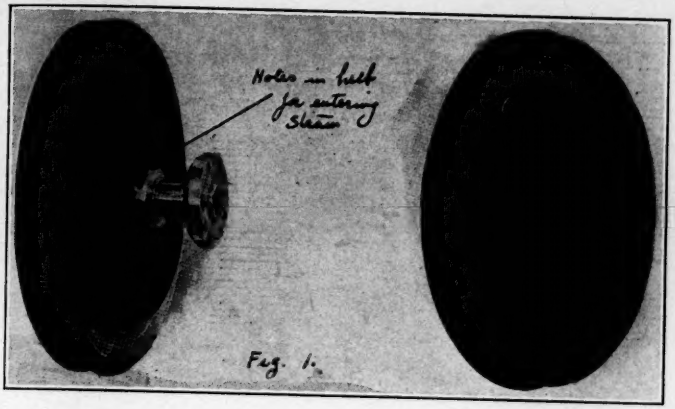
By Lieut. (J. G.) W. B. Carter, U. S. N., Member.

The Ljungström steam turbine which was invented and worked out by Messrs. Birger and Fredrik Ljungström, is a turbine of the radial-flow type, specially adapted to the driving of electric generators, as will be seen later. It is manufactured by the Swedish Ljungström Steam Turbine Co., Ltd., of Friespong, Sweden. The company was founded in 1913, and has taken up the manufacture of turbo-generators in connection with the turbines.

A very superior economy in steam consumption, great reduction in weight and space, combined with increased working safety are the prominent features claimed for the Ljungström turbine system of electric propulsion.

It has been found that in theory the most favorable steam consumption went with the reaction principle. The impulse turbines, however, present less difficulty in utilizing high temperatures and pressures, owing to their mechanical properties. In the Ljungström turbine, which is a reaction turbine, the mechanical difficulties heretofore considered inseparable from the application of the reaction principle have been overcome, it is claimed, and considerable economy from the use of high superheat at high pressure is obtained.

The Ljungström radial-flow turbine has two sets of blade rings, making two turbine wheels which fit together telescopically and rotate in opposite directions. The idea is to place an alternating-current generator on the end of each shaft. The current from these generators is sent to a motor or motors on the propeller shaft or shafts, and the ship thus driven. Up to date no direct-current generators or motors have been used,



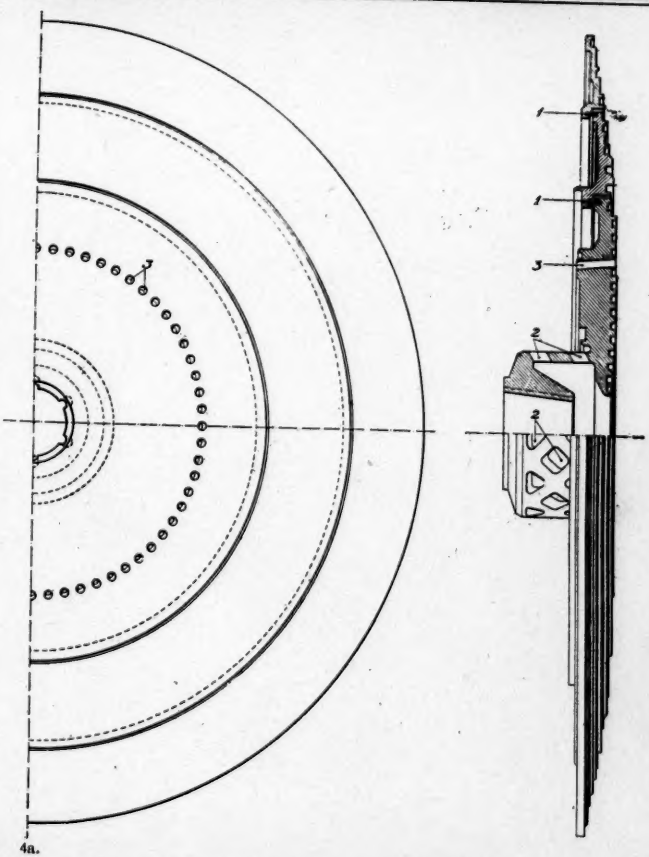


FIG. 2.—CROSS-SECTION OF A TURBINE-DISC.
1. Expansion rings. 2. Steam entrance. 3. Overload parts.

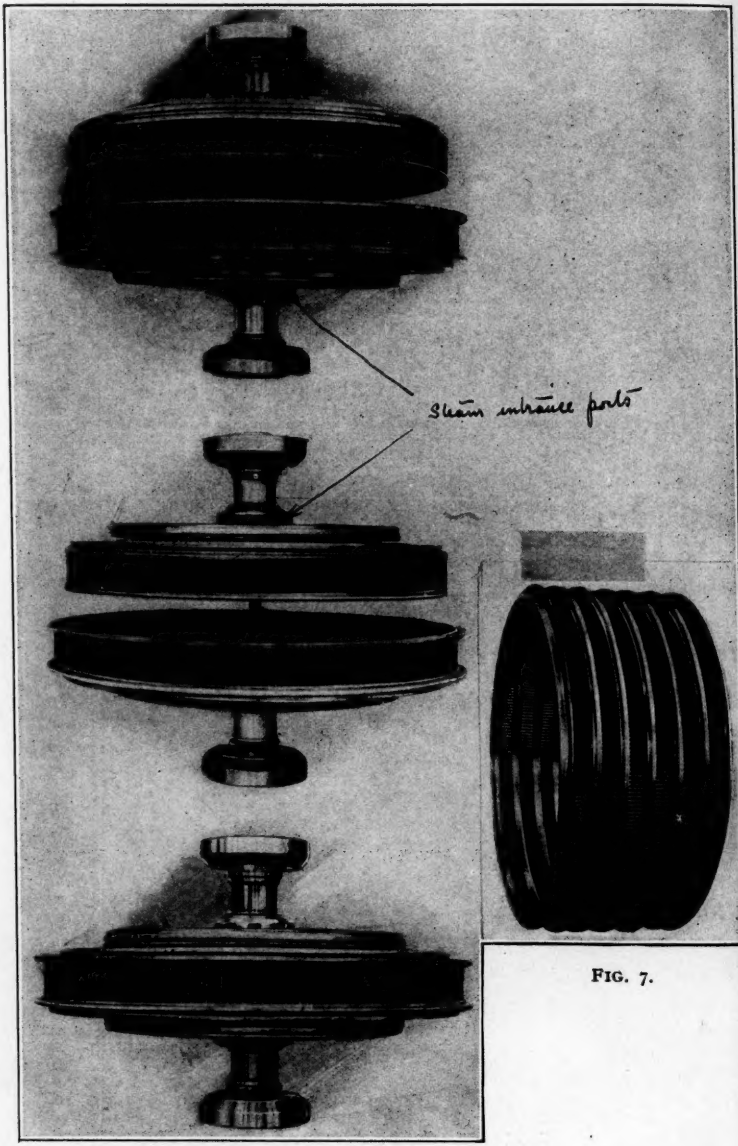


FIG. 6.—COMPLETE BLADE-RING SYSTEM WITH DISCS AND JOURNALS FOR A 1,000 K.W. TURBINE.

FIG. 7.

and therefore this system would not seem to exactly apply for submarines, since the direct-current motors for the propeller shafts are necessary on account of using them with direct current from the storage battery when submerged. However, it is only a step to the direct-current generator or a converter used with the accelerated-current generators, thus making possible the use of direct-current motors on the propeller shafts. In view of this, the Ljungström system is thought worthy of consideration in the case of submarines.

PRINCIPAL PARTS.

Blading.—The turbine consists of two turbine discs with a number of concentrically arranged blade rings fixed on each disc. The blade rings of one disc run between the blade rings of the other disc at the same speed of revolution but in opposite direction. The live steam enters at the center of the turbine and passes through the blade rings radially outward. Although there are no fixed blades, the blades of one disc act as guide blades for the steam going to the blades of the other. It is seen that the relative velocity between blades and guide blades is twice as high as would be the case if one set were stationary, and therefore the work done in each blade ring is four times as great. Consequently, due to the moving guide blades, only one-fourth the number of blade rings will be required to develop the work ordinarily obtained from the steam. This accounts for some of the saving in weight and space claimed.

The steam enters at the center of each turbine disc (discs are practically identical) inside of the innermost blade ring through a number of holes in the hub of the disc. (See Figs. 1 and 2.)

As the steam increases in volume by expansion the length of the turbine blades must be proportioned to meet this increase in volume. The cross-section of a pair of turbine discs with blade rings showing this blade proportioning would appear as shown in Fig. 3.

In turbines of large powers requiring very large quantities of steam, single blades in the wide rings would not have suf-

ficient tensile strength to withstand the stresses set up by the centrifugal force at high speeds of rotation. In order to get around this difficulty the widest blade rings are divided into sections shown in Fig. 4. Something like Fig. 5 would result in large turbines if full-length blades were used. In Fig. 4 the sections are connected to strengthening rings which are attached to the outer edges of the blade rings.

FIG. 3.

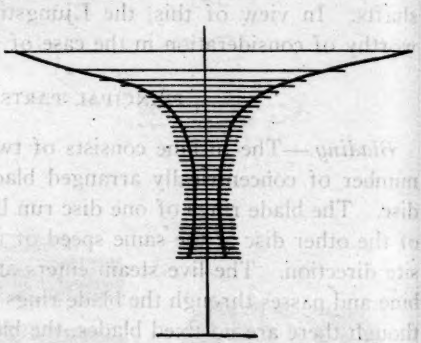


FIG. 5.

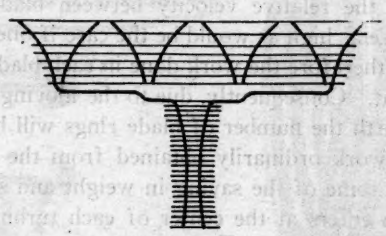


FIG. 4.

The manufacturers feature their design and workmanship of blade rings and blades. Fig. 6 shows the complete blade-ring system for a 1,000-k.w. turbine. It will be observed that this has the continuous full-length blades. Fig. 7 shows a blade ring consisting of several sections intended for a larger turbine.

The method of assembling blades and rings is shown by Fig. 8. The nomenclature is as follows:

Turbine disc	1	Tightening strips	7
Seating ring	2	Caulking strips	8
Caulking strip	3	Strengthening rings	9
Expansion ring	4	Dovetail profile	10
Rolling ridges	5 and 6	Turbine blades	11

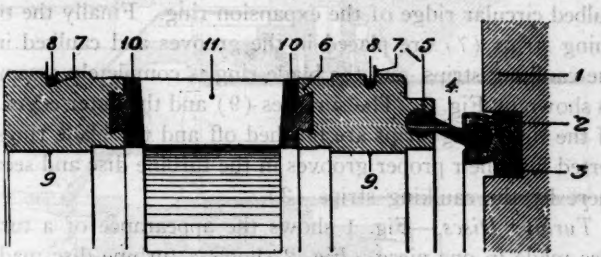


FIG. 8.—SECTION SHOWING BLADE RINGS AND STRENGTHENING RINGS.

The ends of the blades are given the proper shape to fit into punched slots and are then welded to the disc by spot welding. During the welding the discs are held on a mandrel and the blades are kept in their proper positions by means of thin sheet-iron strips in which are punched holes of the same cross-section as the blades, and at a distance corresponding to the pitch and spacing of the blades. These strips are put in place at the time of assembling the blades and rings, and the angles of the blades are thus adjusted, and the blades prevented from being thrown out of place during welding. When the welding is completed the sheet-iron strips are cut away and the welding discs turned down to a finish of the profile shown by 10 in Fig. 8. Outside these welding rings strengthening rings are fastened. The strengthening rings are first given a cross-section shape as shown by 12 of Fig. 9, then the projecting edges (6)

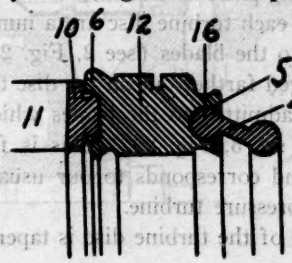


FIG. 9.

are rolled down to fit over the dove-tailed rim of the welding ring. The expansion ring is also secured by rolling down the edges (5) on the outer side of the strengthening ring. The seating-ring (2) edges are likewise rolled down over the outer bulbed circular ridge of the expansion ring. Finally the tightening strips (7) are placed in the grooves and caulked in by the caulking strips, and the blade ring is completely assembled as shown by Fig. 8. The surfaces (9) and the outer edges (7) of the tightening strips are finished off and the blade rings inserted into their proper grooves in the turbine disc and secured there by the caulking strips (3).

Turbine Discs.—Fig. 1 shows the appearance of a turbine disc made in one piece. Fig. 2 shows a turbine disc made in several sections. It is made in the latter way to avoid the stresses and alterations in shape due to the influence of the varying and irregularly distributed steam temperatures and pressures. This occurs most frequently when starting up. It is when steam is first admitted to the turbine that variations in pressure, temperature and load are most likely to occur. Even under uniform running there would be high pressure, high-temperature steam in contact with the center of the disc, while at the circumference or outer blades the steam would be at low pressure and lower temperature. Under such conditions a plane disc would take a dished shape. A very thick disc which would not dish would be subjected to internal stresses which would be liable to crack the material. The sections shown in disc of Fig. 2 are joined by means of expansion rings, which allow for any expansion or contraction and prevent stresses from taking place in the material.

In each hub of each turbine disc are a number of holes for admitting steam to the blades (see 2, Fig. 2); also there are some holes arranged farther out on the disc to allow full-pressure steam to be admitted to the blades which ordinarily use expanded steam (see 3, Fig. 2). This is for full-power or overload work, and corresponds to our usual practice of by-passing the high-pressure turbine.

The center hole of the turbine disc is tapering, and the hub

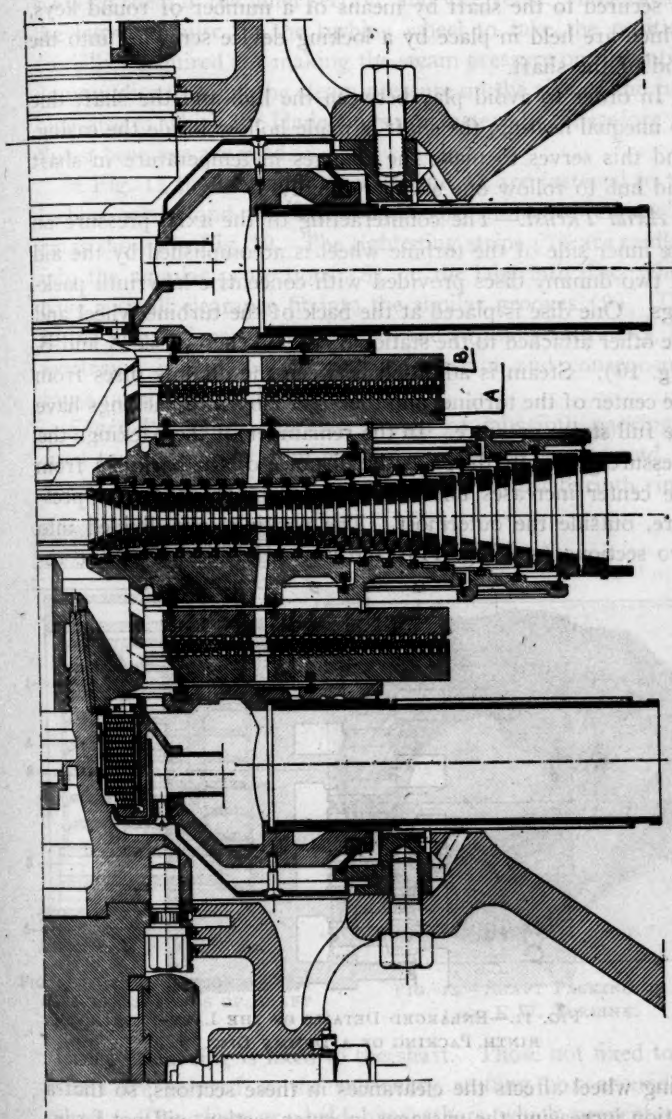


FIG. 10.—LONGITUDINAL SECTION OF 1,000 K.W. TURBINE, LOWER PART.

is secured to the shaft by means of a number of round keys, which are held in place by a locking device screwing into the end of the shaft.

In order to avoid play between the hub and the shaft due to unequal heating, the shaft is made hollow inside the casing, and this serves to make the changes in temperature in shaft and hub to follow one another rapidly.

Axial Thrust.—The counteracting of the axial pressure on the inner side of the turbine wheel is accomplished by the aid of two dummy discs provided with concentric labyrinth packings. One disc is placed at the back of the turbine wheel and the other attached to the stationary steam chests (see A and B, Fig. 10). Steam is admitted between the dummy discs from the center of the turbine, and the inner labyrinth packings have the full steam pressure. In the remainder of the packings the pressure gradually drops as the distance of the packings from the center increases until it finally drops to condenser pressure, outside the outermost. The packings are divided into two sections (see Fig. 11), and any axial motion of the ro-

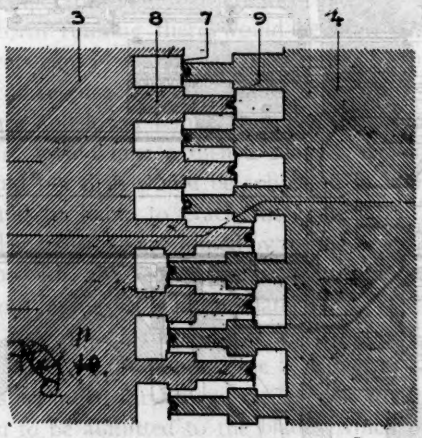


FIG. 11.—ENLARGED DETAILS OF THE LABYRINTH PACKING OF A DUMMY DISC.

tating wheel affects the clearances in these sections, so that a motion increasing the passages in outer section will not be in-

creasing them in the inner section, and vice versa. This causes the labyrinth disc on the turbine wheel to take the position (axially) required for making the steam pressure on the entire dummy disc to equal the steam pressure on the side of the turbine disc to which the blade rings are fastened. Therefore no thrust bearings are used.

In Fig. 11 the labyrinth discs (3 and 4) are fastened to the turbine wheel and the steam chests by means of expansion rings shown in Fig. 10. The tightening strips (7) are caulked into the annular projections (8) on the labyrinth disc, which have a small clearance fit into the similar grooves (9). Due to the proportions of the metal in the two labyrinth discs, the expansion is expected to remain constant and consequently cause no change in the clearances.

Shaft Packings.—Another design of labyrinth packing is used to prevent air from getting into the turbine around the shaft. Fig. 13 shows the cross-section of two labyrinth rings.

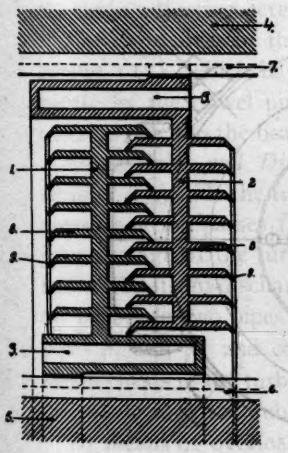


FIG. 13.—CROSS-SECTION OF TWO LABYRINTH RINGS OF SHAFT PACKING.

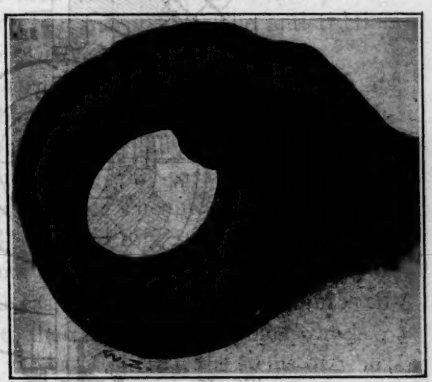


FIG. 14.—SHAFT PACKING OF A 1,000 K.W. TURBINE.

Every other ring is fixed to the shaft. Those not fixed to the shaft are secured to the stationary stuffing box around the shaft. The rings are held by feathers which prevent them

from turning. The packing edges of the rings are very thin and are set at an angle of 45 degrees, and if by accident these edges come in contact with the ribs of the next ring there will be no damage done, the edge merely wearing off until contact ceases.

The thickness of all the rings is the same, and therefore the clearances between ribs is expected to be constant, either hot or cold, as the expansion and contraction will remain constant in each. The clearances are therefore made very small, and consequently the leakage is expected to be very small. There are special pipes fitted to the packing boxes so that leaking steam may be sent to the feed-water heater and some of its heat utilized.

Main Bearings.—A cross-section of the bearing with its adjusting screws is shown by Fig. 15. The bearing consists of

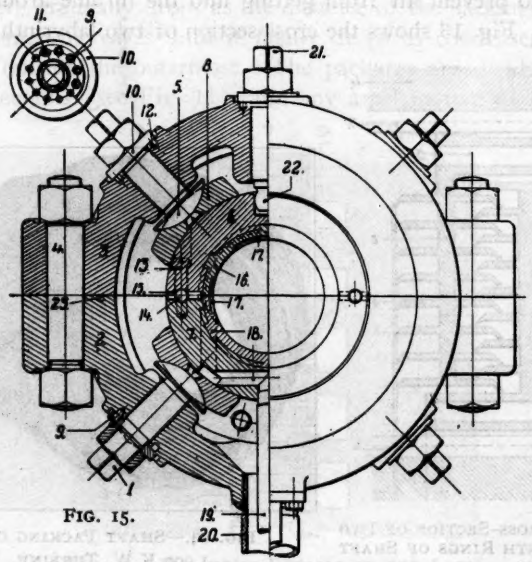


FIG. 15.

two halves (6) and (7) within bearing boxes. The halves rest against adjusting screws through intermediary washers (8). The adjusting screws are locked by means of set screws

(9) and locking washers (10). There are twelve holes in the washer and only eleven in the bearing surface on which it rests. This allows for very accurate adjustment of bearing without too fine a threading of adjusting screws.

The bolts (4) which hold the upper and lower halves of the bearing boxes together are tapered and fit accurately in reamed holes, thus at the same time serving as securing dowels for the cap. After the bearing boxes are adjusted, which is usually made during erection of the turbine in the shop, they will keep that position accurately and never need to be changed. To prevent the bearing from slipping axially and from being turned by friction another adjusting screw (21) is used. It has a spherical end which fits into a corresponding recess in the upper half of the bearing. The spherical end is made eccentrically to the axis of the adjusting screw, and therefore the bearing can be shifted axially by turning the screw, which is afterward locked by a washer and set screw in the same way the main adjusting screws are locked.

The two halves of the bearing are fastened together by the screws (13), which have their relative position fixed permanently by the dowel pins (14) and (15). The anti-friction metal is put into the bearing halves in the usual manner.

Steam Inlet and Discharge.—The throttle valve with its strainer is outside the turbine casing (see Fig. 19). The high-pressure steam passes from it horizontally through the steam-inlet passage of the turbine casing, this passage being located within the steam-discharge passage. The inlet passage branches off here into two pipes B_1 and B_2 (Fig. 16), which join steel pipes b_1 and b_2, and communicate with the chambers of the steam chests in the turbine casing. From here the steam is admitted through the hubs to the center of the turbine wheels.

By means of overload valves (Fig. 12), the stems of which are accessible outside of turbine casing, steam can be admitted when desired into the overload channels (Figs. 2, 11, 12). The overload valves are constructed to act automatically when conditions necessitate. The exhaust steam passes from the outermost blade ring through the annular space between it and

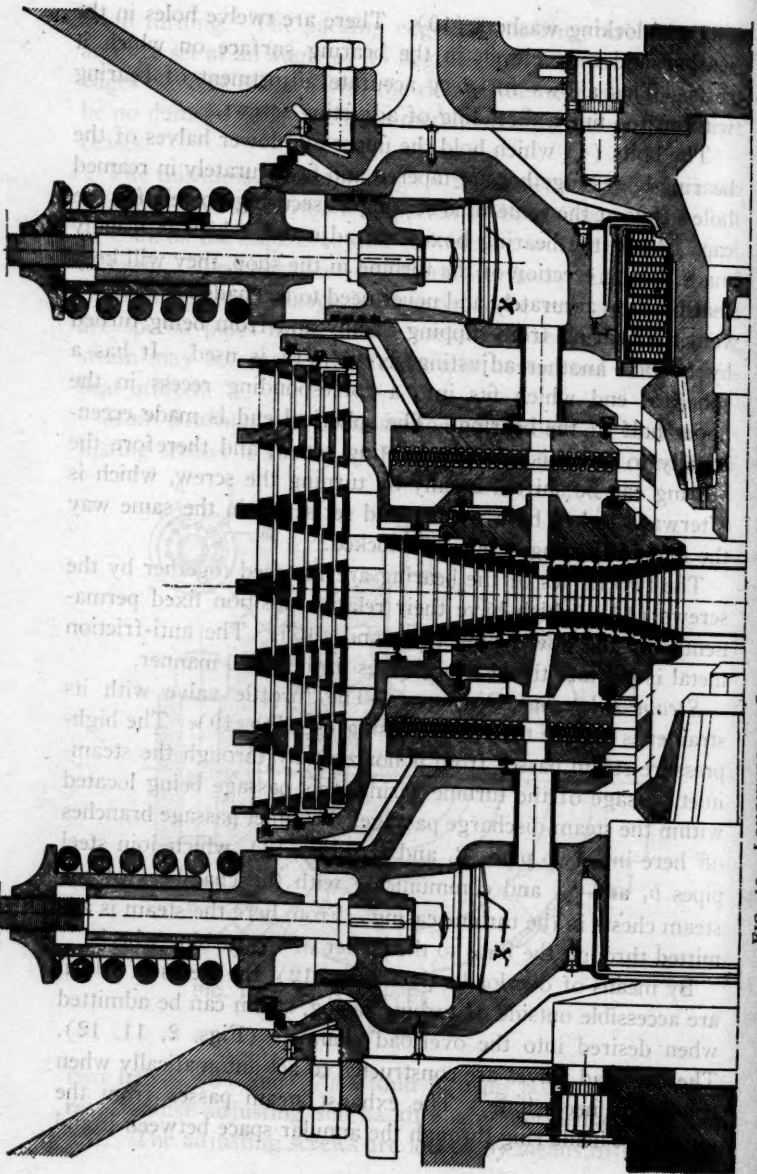


FIG. 12.—LONGITUDINAL SECTION OF 5,000 K.W. TURBINE, UPPER PART.
X = Overload valves.

the turbine casing and through the discharge passage into the condenser below.

HEAT RADIATION INSULATION.

Since the turbine is located in the exhaust passage and fitted with the small cross-section area expansion rings hereinbefore mentioned, the conduction of heat from the hot high-pressure parts to the comparatively cool low-pressure parts and the casing is very slow and the casing requires no lagging. The only parts which do require heat insulation are the throttle valve and strainer and the pipes carrying the high-pressure steam through the exhaust passage to the center of the turbine wheel, and the small annular steam chests located in the turbine casing at the outer sides of the turbine wheels. As before mentioned, the shafts are hollow, and the small amount of heat transmitted from the turbine to the generators is absorbed by the lubricating oil circulating in the bearings. The steam piping of the installation must, of course, be lagged as usual, in fact rather more than usual, as the steam used may be above 700 degrees F. (see test given later on). A considerable advantage for ships in the tropics is claimed for the Ljungström system, as this radiation factor is low and the engine room may be kept cooler than otherwise.

Lubrication.—The lubrication is by means of the usual rotary-gear oil pump driven off the vertical governor shaft by worm gearing. The pump is mounted in the oil tank together with an oil cooler and a hand oil pump (see Fig. 19a). In Fig. 17 the oil-pressure pipe (32) leading to the bearings and the return pipe (33) is shown. The connection of these pipes to the bearings is shown at the bearing (6) and in Fig. 15, where the pressure oil pipe to the box is designated by (19). The oil is pumped out (19) and into channels (18) drilled in lower bearing. Used oil flows from bearing box to reservoir through pipe (20). The method of preventing leakage of oil from the boxes is shown in Fig. 16. The body of the box surrounding the bearing is provided with two grooves (*c*) and (*d*), and with narrow ribs (*e*), the edges of which close tightly on the

FIG. 12.—LONGITUDINAL SECTION OF 5,000 K.W. TURBINE, UPPER PART.
x — Overload valves.

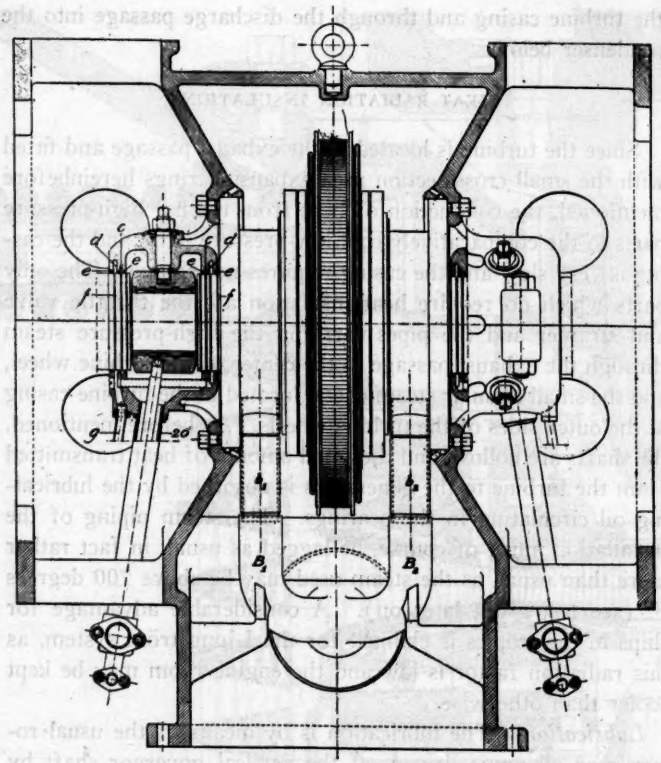


FIG. 16.—LONGITUDINAL SECTION OF 1,000 K.W. TURBINE.

shaft. In the shaft, outside the journal, is turned a groove (f), the edges of which throw off the oil. From groove (d) is led a drain pipe (g) for used oil. The groove (c) communicates with discharge pipe 20 and with the narrow oil passage 23 of Fig. 15, which is designed to prevent leakage between lower and upper half of bearing box.

The quantity of oil per minute for a 1,000-k.w. turbine is about 6.6 gallons per bearing box. The pump for this would have a capacity of about 27 gallons per minute at about 35 pounds per square inch pressure. Only about half this pres-

sure is required for bearings, cooler and frictional resistance of piping, but the higher pressure is to permit of operating the throttle valve by an oil cylinder.

GOVERNOR.

The governor is a centrifugal one mounted on the same vertical shaft as the oil pump. As mentioned above, this shaft is worm driven from the turbine shaft. The motion of the governor sleeve is transmitted by means of the lever (37) (see Figs. 17, 18 and 19) to the link (38) reaching down to the oil reservoir, and joining there to a small slide valve by means of which the oil pressure is controlled. From this control valve the oil is carried through a pipe (39) to the oil cylinder (40) arranged on the throttle valve (41). The oil pressure acts on the bottom side of the piston in the cylinder, the piston rod (50) of which is connected to the spindle of the throttle valve. The top side of the piston is acted on by a powerful spring. Underneath the valve a water separator and steam strainer are arranged. The control valve is opened by the oil pressure below the piston and is closed by the spiral spring as soon as the oil pressure drops below a certain limit. The position of the slide valve in the oil reservoir, which valve is directly actuated by the governor, determines the pressure of oil in the cylinder over the throttle valve and thus the position of the oil piston and valve spindle. The valve spindle is connected to the levers (42) and (44) and to the shaft (43), whereby return motion is given to the sliding sleeve surrounding the slide valve in the oil tank, so that the motion of the throttle-valve spindle corresponds to that of the governor sleeve.

Above the oil-pressure piston of the throttle valve is a spiral spring set at from 600 to 800 pounds, *i. e.*, about double the force available through the return motion on the governor sleeve. When the oil pressure drops to zero the spring will close the throttle valve. This will occur if the oil pressure is released by hand when stopping, or if the emergency governor takes action.

An emergency governor is fitted to each of the main shafts of the turbine outside the bearing boxes (see Fig. 17). It acts by means of a releasing device or trigger on a slide valve, which cuts off the oil pressure from the oil cylinder of the gov-

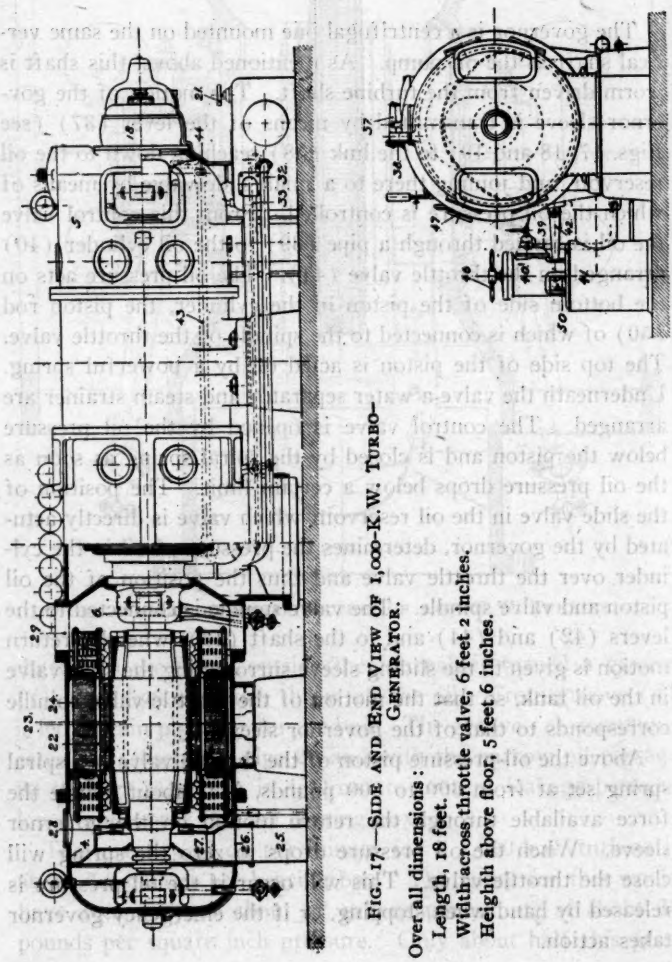


FIG. 17.—SIDE AND END VIEW OF 1,000-K.W. TURBO-GENERATOR.

Over all dimensions:
Length, 18 feet.
Width across throttle valve, 6 feet 2 inches.
Height above floor, 5 feet 6 inches.

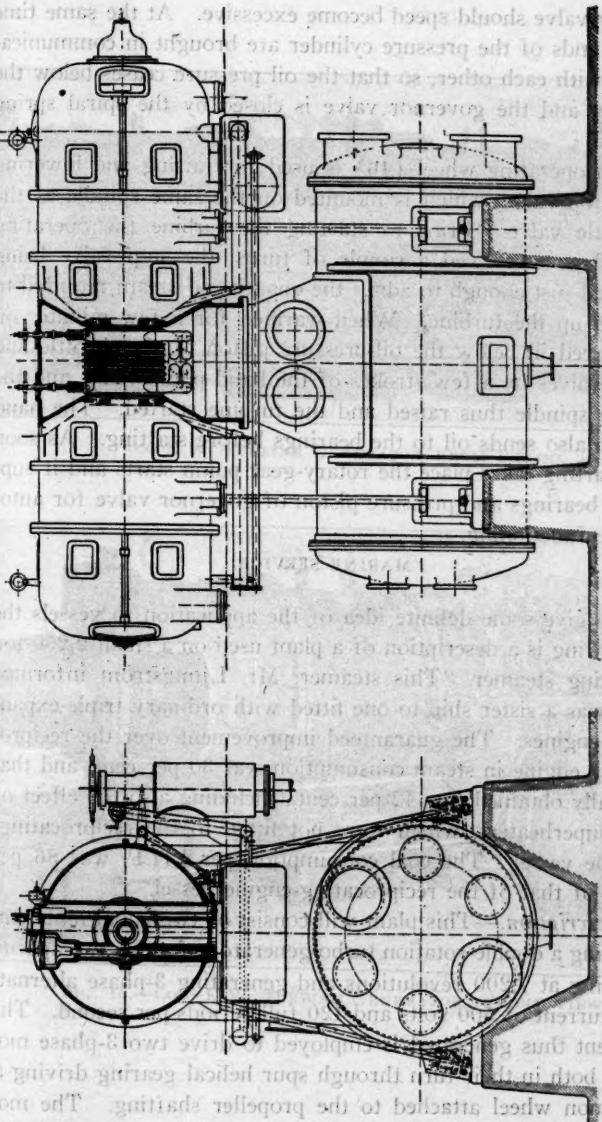


FIG. 18.—END AND SIDE VIEW OF 5,000 K.W. TURBO-GENERATOR AND CONDENSER.
Over-all dimensions:—Length, 23 feet 4 inches. Width across throttle valve, 10 feet. Height, including condenser, 21 feet.

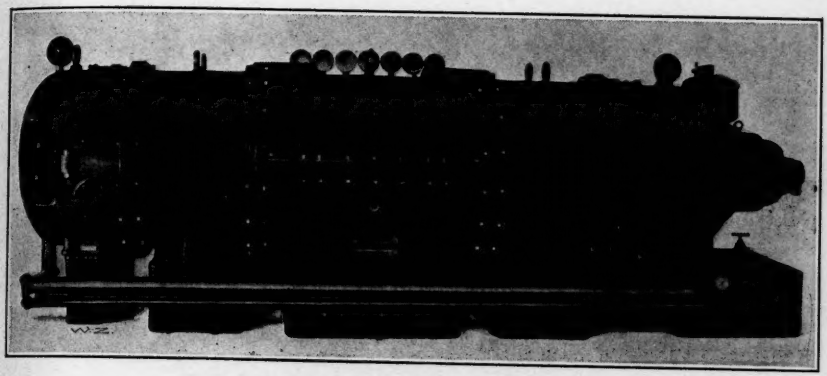
ernor valve should speed become excessive. At the same time both ends of the pressure cylinder are brought in communication with each other, so that the oil pressure ceases below the piston and the governor valve is closed by the spiral spring above.

An operating wheel (46) is used for raising and lowering the stop valve, which is mounted on the same spindle as the throttle valve. Prior to starting the turbine the operating wheel is unscrewed a couple of turns, the stop valve being opened just enough to admit the quantity of steam required to warm up the turbine. When warmed for a few minutes oil is forced in below the oil-pressure piston of the throttle and stop valves by a few strokes of the hand pump (21), and the valve spindle thus raised and the turbine started. The hand pump also sends oil to the bearings before starting. As soon as starting takes place the rotary-gear pump starts and it supplies bearings and pressure piston of governor valve for automatic functioning.

MARINE SERVICE.

To give some definite idea of the application to vessels the following is a description of a plant used on a small 2,250-ton coasting steamer. This steamer, Mr. Ljungström informed me, was a sister ship to one fitted with ordinary triple-expansion engines. The guaranteed improvement over the reciprocating engine in steam consumption was 30 per cent. and that actually obtained was 42 per cent., including also the effect of the superheaters, which were not fitted to the reciprocating-engine vessel. The coal consumption per I.H.P. was 86 per cent. of that of the reciprocating-engine vessel.

Description.—This plant will consist of two equal units embodying a double-rotation turbo-generator of 400 k.w. capacity running at 7,200 revolutions and generating 3-phase alternating current of 500 volts and 120 full periods per second. The current thus generated is employed to drive two 3-phase motors, both in their turn through spur helical gearing driving a common wheel attached to the propeller shafting. The mo-



1,000 K.W. Turbine Generator From Behind.

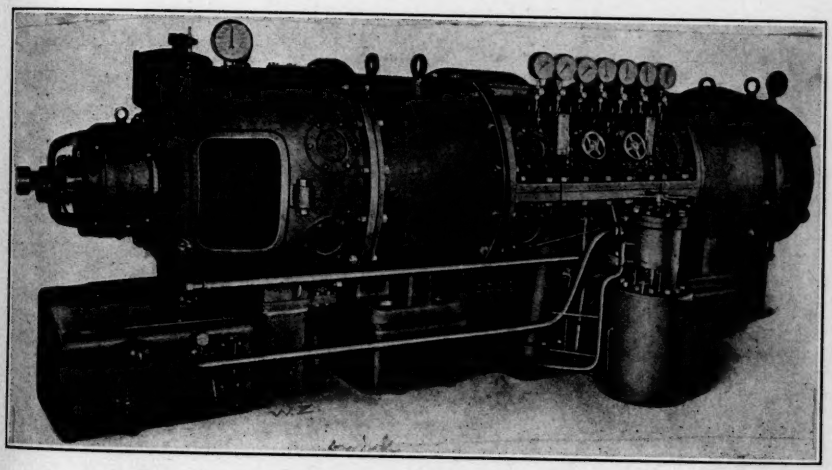


Fig. 19.—1,000 K.W. Turbine Generator; Front View.

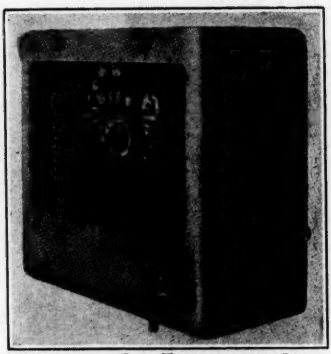


FIG. 19A.—OIL-TANK WITH OIL-COOLER, GEAR-PUMP, HAND-PUMP, AND PRESSURE-REGULATING VALVE.

tors will run at a speed of 900 revolutions and the propeller shafting at 90 revolutions per minute.

"Each turbo-generator unit, consisting of turbine and two electric generators, is mounted directly on top of an ordinary surface condenser, having fitted on it electrically-driven air, circulating and feed pumps.

"The air pump is to be of the 'rotary' kinetic type and the two others of the 'centrifugal' type.

"The condenser is of the ordinary marine type, and has an ample cooling surface, enabling a vacuum of about 95 per cent. to be kept up when running at full speed. The above-mentioned pumps are to be arranged so as to enable them to be used for either or for both of the generator sets, thus minimizing the chance of total break-down. The bilge pumps are driven independently by steam.

"The maneuvering is done electrically. Speed regulation is obtained by resistance in the motor circuit and the reversal is effected by interchanging the poles of the phases. When the generators are unloaded, for one reason or the other, an automatic governor shuts off the steam accordingly.

"The thrust from the propeller is taken up by a ball bearing at the forward end of the shaft carrying the common spur wheel, but in addition to this there is also a thrust bearing of the usual horse-shoe type, which comes in action if anything should happen to the ball bearing.

"All lubrication of the moving parts of the generators and their auxiliaries, as well as the helical gear transmission, is automatic.

"The steam for the turbo-generators is supplied by two boilers of the usual cylindrical type at a pressure of 220 pounds above the atmosphere and at a temperature of 350 degrees Celsius.

"The boilers are 10 feet 10 inches diameter, and 10 feet 9 inches long, having a total heating surface (exclusive of the superheater) of c:a 2,140 square feet and a total bar surface of c:a 57 square feet. Each boiler has two corrugated furnaces 35 inches internal diameter.

"The boilers are fitted with Howden's draft and with Schmidt's superheating arrangement. The air for Howden's draft is supplied by the generators, the ventilation air from the rotors being carried by means of air ducts to this intake, where it is heated in the usual way before being admitted to the furnaces.

"The superheater has a total surface of c: a 770 square feet divided into 48 sections.

"The boilers also have to supply saturated steam for the steering gear, electric-light installation, ballast, feed and ash-ejection donkies and deck machinery.

"The feed water is taken through heaters and filters before being returned to the boilers.

"In order to protect the main condensers from fat and dirt from the auxiliary steam-driven donkies and the deck machinery, an auxiliary or winch condenser is installed with all necessary connections."

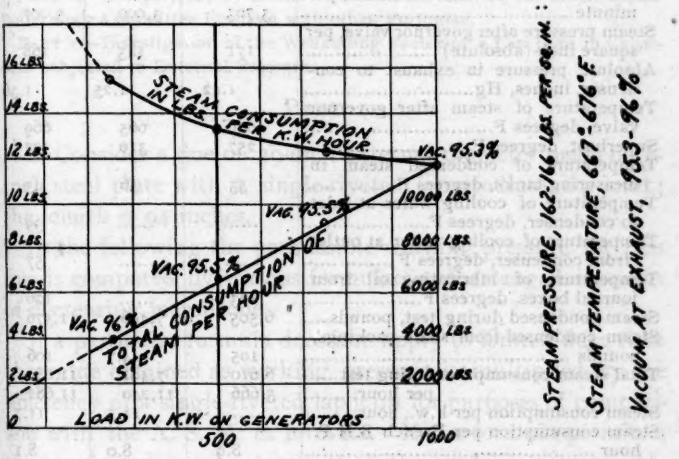
Tests of two turbines are given. They are taken from trade catalogue. (The 500-k.w. is assumed to be one of those installed in the above-mentioned vessel, but the writer has no proof that such is the case.)

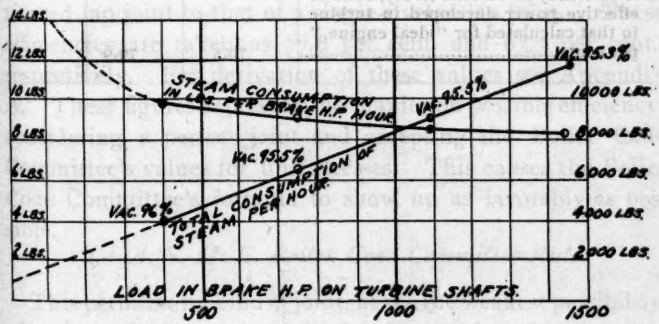
The table and curves gives a summary of the results obtained in some official steam-consumption and efficiency trials made in Stockholm in 1910 and 1911 on two steam turbines of the Ljungström type designed for a normal output of 750 and 1,500 B.H.P. respectively (corresponding to 500 and 1,000 k.w.).

The test recorded in the last column was carried on at full load, and also at ¾, ½ and ¼ load. The curves reproduced show the results of this test both with regard to the total steam consumption of the turbine per hour and as regards the steam consumption per k.w. and per B.H.P.

The improvements made in design since the above-mentioned tests have, it is claimed, further reduced the steam consumption, which will be lower still for units of greater output than in the case of the 1,000 k.w. turbine generator just mentioned.

According to calculations, a turbine generator designed for a normal output of 5,000 k.w. will, at a steam pressure of 300 pounds per square inch, a steam temperature of 750 degrees F., and an absolute back pressure in the steam exhaust of 1 inch of Hg., consume about 6.5 pounds of steam per B.H.P. per hour.





Official Tests of Steam Turbines With Two Shafts of the Ljungström Double Rotation Type.

Rated output of steam turbine	750 B.H.P.	1,500 B.H.P.	
Date of test	Nov., 1910	Oct. and Dec., 1911.	
Duration, minutes	70	40	60
Total load on both turbine shafts, French B.H.P.	635.7	1,405	1,439.7
Electric output of generators, k.w.	.....	972	1,000.5
Revolutions of both turbine shafts, per minute	3,703	3,000	3,007
Steam pressure after governor valve, per square inch (absolute)	171	143	168
Absolute pressure in exhaust to condenser, inches, Hg	1.42	1.25	1.36
Temperature of steam after governor valve, degrees F	626	665	669
Superheat, degrees F	257	310	302
Temperature of condensed steam in measuring tanks, degrees F	53	56	43
Temperature of cooling water at inlet to condenser, degrees F	.....	.....	39
Temperature of cooling water at outlet from condenser, degrees F	.....	.....	57
Temperature of lubricating oil from journal boxes, degrees F	123	.....	126
Steam condensed during test, pounds	6,505	7,446	11,576
Steam condensed from shaft packings, pounds	105	44.7	106
Total steam consumption during test	6,610	7,491	11,682
per hour	5,666	11,240	11,682
Steam consumption per k.w., hour	.....	11.5	11.7
Steam consumption per French B.H.P. hour	8.9	8.0	8.1
Consumption per French B.H.P. per hour, "ideal engine" (computed)	6.28	6.15	6.05
Net efficiency of turbine (*i. e.*), ratio of effective power developed in turbine to that calculated for "ideal engine," per cent	70.5	76.9	74.6

STRENGTH OF BOILER FURNACES.

By John Airey,
Assistant Professor of Engineering Mechanics, University of Michigan.

PART 1.—Comparison, by means of a concrete case, of the A. S. M. E. Boiler Code Committee Formula with other Formulae.

PART 2.—Investigation of the Weakening Factor of a Lap Joint in Cylinders subjected to External Pressure.

PART 1.

1. Consider a flue of 29-inch external diameter made of ½-inch steel plate with a single-riveted longitudinal lap joint; the length is 94 inches.

In the following the permissible working pressure for this flue is computed by various formulae and arranged in order of decreasing magnitude.

If a particular formula does not apply to *jointed* flues then the value obtained as working pressure is multiplied by the efficiency of a single-riveted lap joint for purposes of comparison with the A. S. M. E. formula. If the formula permits a double-riveted lap joint, say, then the working pressure obtained is reduced in the proportion of the efficiency of a double-riveted lap joint to that of a single-riveted lap joint. These efficiencies are taken as 77.6 per cent. and 63.3 per cent., respectively. For derivation of these values see Appendix A. These figures represent the maximum possible efficiency, considering a perfect joint and accepting the Boiler Code Committee's values for unit stresses. This causes the Boiler Code Committee's formula to show up as favorably as possible.

2.—*A. S. M. E. Boiler Code Committee Rule.*

This permits any kind of joint, hence the weakest possibility, viz: a single-riveted lap joint is taken in the other cases for purpose of comparison. The efficiency of the joint is, however, always taken as the maximum possible, as stated in Par. 1.

In the case considered, as the length exceeds 120 times the thickness, the second formula given by the Boiler Code Committee must be used, viz:

$$P = \frac{4{,}250T^2}{LD},$$

where

P = maximum allowable working pressure, pounds per square inch;
D = outside diameter of furnace in inches;
L = length of furnace in inches;
T = thickness of furnace wall in sixteenths of an inch;

$$P = \frac{4{,}250 \times 8^2}{94 \times 29} = 99.8 \text{ pounds per square inch.}$$

3.—*Fairbairn's Rule.*

This formula does not allow for joint. Fairbairn's rule, although the parent formula, is recognized as giving working pressures far too high if the present-day factor of safety of 5 be employed.

$$p = \frac{806{,}300 \times t^{2.19}}{LD},$$

where

t = thickness of tube in inches;
L = length of tube in feet;
D = external diameter in inches;
p = collapsing pressure in pounds per square inch;

$$p = \frac{806{,}300 \times .5^{2.19}}{\frac{94}{12} \times 29} = 777 \text{ pounds per square inch.}$$

Allowing a factor of safety of five (see Progress Report of Boiler Code Committee, A. S. M. E.) gives 155.4 pounds per square inch.

Taking 63.3 per cent. of this to allow for a single-riveted lap joint makes the value of working pressure for purpose of comparison with the A. S. M. E. result to be 98.5 pounds per square inch, or practically the same as the value given by the A. S. M. E. formula.

4.—*Modified Fairbairn Rule.*

The working pressure of a tube subjected to external pressure will obviously not be inversely proportional to the length indefinitely, as Fairbairn's rule would indicate. This has been partially counteracted by using a variable factor of safety $\sqrt{\frac{300}{L}}$, where L is length in feet.

For the concrete case under discussion this gives working pressure:

$$= \frac{777}{\sqrt{\frac{300 \times 12}{94}}} = 125.8 \text{ pounds per square inch.}$$

Making allowance for the lap joint gives $125.8 \times .633 = 79.8$ pounds per square inch, or 80.0 per cent. of the value given by the A. S. M. E. formula.

5.—*Bureau Veritas Rule.*

This formula makes a distinction between the case where the flue is truly circular (*i.e.* butt joints or lapped and bevelled), and where the flue is not circular. It is worth while noting in passing that the difference in the constant given is by no means as great as it should be when viewed in connection with the subject matter of Part 2 of this discussion.

$$P = \frac{Ct^2}{DL},$$

where

t = thickness of plate in inches;
D = outside diameter in inches;
P = working pressure in lb. per sq. in. above atmosphere;
L = length of furnace in feet, or length between rings;
C = 70,000 when furnace is truly circular and the longitudinal seams are welded, butted, or lapped and bevelled, and double riveted in last named case;
C = 60,000 when the furnace is not truly circular, or when the longitudinal seams are simply lapped.

Above constants are for iron of fair quality. Multiply constants by 1.2 for mild steel.

For the concrete case under discussion this gives

$$P = \frac{60{,}000 \times 1.2 \times .5^2}{29 \times \frac{94}{12}} = 79.3 \text{ pounds per square inch, or } 79.4$$

per cent. of the value given by the A. S. M. E. formula.

The above formula gives a much stronger design for the truly circular case than it does for the not truly circular case.

6.—*Hutton's Rule.*

This does not allow for a joint.

$$P = \frac{Ct^2}{dL}$$

where

L = length in inches;
d = outside diameter in inches;
t = thickness in thirty-seconds of an inch;
C = 660 for steel;
p = collapsing pressure in pounds per square inch.

For the concrete case under discussion this gives

$$p = \frac{660 \times 16^2}{29 \times 94} = 602 \text{ pounds per square inch.}$$

Allowing a factor of safety of five and reducing to allow for single-riveted lap joint gives $\frac{.633 \times 602}{5} = 76.2$ pounds per square inch, or 76.4 per cent. of the value given by the A. S. M. E. formula.

7.—*German Government Rule.*

$$\text{Thickness} = K \times P \times L \times D + .2 \text{ cm.},$$

where

P = working pressure in kilo per square cm.;
L = length between stiffeners in cms.;
D = external diameter of furnace in cms.;
K = .0035 when the longitudinal joints are lapped only.

For the concrete case under discussion this gives

$$2.54 \times .5 = .0035 \frac{P \times .4536}{2.54^2} \times 94 \times 2.54 \times 29 \times 2.54 + .2.$$

$$P = \frac{(1.27 - .2)^2}{.0035^2} \times \frac{1}{.4536 \times 94 \times 29} = 75.6 \text{ pounds per sq. inch, or } 75.7 \text{ per cent. of the value given by the A. S. M. E. formula.}$$

8.—*Lloyd's Rule.*

This does not allow for joint.

$$P = \frac{89{,}600t^2}{LD},$$

where

P = working pressure in pounds per square inch;
t = thickness of plate in inches;
L = length of furnace in feet;
D = outside diameter of furnace in inches.

For the concrete case under discussion this gives

$$P = \frac{89{,}600 \times .5^2}{\frac{94}{12} \times 29} = 98.7 \text{ pounds per square inch.}$$

Reducing this to the corresponding value for single-riveted lap joint gives $98.7 \times .633 = 62.4$ pounds per square inch, or 62.5 per cent. of the value given by the A. S. M. E. formula.

9.—*U. S. Statute Rule is Same as Lloyd's Rule.*

10.—*Michael Longridge's Rule.*

Michael Longridge takes C in Hutton's formula as 540 instead of 660: this gives $\frac{540}{660}$ of the value obtained in Par. 6 with Hutton's formula, or $\frac{540}{660} \times 76.2 = 62.4$ pounds per square inch, or 62.5 per cent. of the value given by the A. S. M. E. formula.

11.—*British Board of Trade Rule.*

This permits a double-riveted lap joint.

$$p = \frac{75{,}000t^2}{(L + 1)D},$$

where

p = working pressure in pounds per square inch;
t = thickness in inches;
L = length in feet;
D = external diameter in inches.

For the concrete case under discussion this gives

$$p = \frac{75{,}000 \times .5^2}{\left(\frac{94}{12} + 1\right) 29} = 73 \text{ pounds per square inch.}$$

Reducing this to the corresponding value for a single-riveted lap joint gives $73 \times \frac{.633}{.776} = 59.6$ pounds per square inch, or 59.7 per cent. of the value given by the A. S. M. E. formula.

12. The previous results are collected below to facilitate comparison:

	Working pressure. Pounds per sq. inch.	Percentage.
A. S. M. E. Boiler Code Committee,	99.8	100.0
Fairbairn,	98.5	98.7
Fairbairn (modified),	79.8	80.0
Bureau Veritas,	79.3	79.4
Hutton,	76.2	76.4
German Government,	75.6	75.7
Lloyd's,	62.4	62.5
U. S. Statute,	62.4	62.5
Michael Longridge,	62.4	62.5
British Board of Trade,	59.6	59.7

PART 2.

INVESTIGATION OF THE WEAKENING FACTOR OF A LONGITUDINAL LAP JOINT IN A CYLINDER SUBJECTED TO EXTERNAL PRESSURE.

13. The conventional engineers' theory on the strength of a lap joint assumes the stress uniformly distributed over the cross section, either in the solid plate or between rivets, and ignores eccentricity of loading due to plates being out of line. This theory is set forth in the accompanying appendix because the results obtained, using the Boiler Code Committee's allowable stresses, have been quoted in Part 1.

Probably none would dispute the fact that a lap joint is considerably weaker than it is supposed to be, due to this neglected eccentricity of loading, but just how much weaker is considered debatable.

The following is a solution of the weakening effect. The analysis is devoid of the very uncertain assumptions unavoidable in the study of the joint itself, *i. e.*, the extent of friction between the plates, the uniformity of distribution of stress between rivets, bearing, shear, etc. The reason for this is that when a flue fails it invariably buckles, the plate bending most just where the lapping ends. This analysis shows that the weakness here is considerably greater than that of a joint proper and the weakness is inherent in the lapping, a double-riveted lap joint being merely slightly less bad than a single-riveted one. It will be shown that lack of circularity is the dangerous feature and that a single-strap butt joint may be a great deal stronger construction than a lap joint, although the conventional theory makes them identically equal—a single-strap butt joint being looked upon merely as two lap joints.

The customary engineers' theory for distribution of stress due to eccentric loading is used in the discussion and hence, for convenience, will be recapitulated here.

14. If the cross-section of a body is rectangular, as in Fig. 1, dimensions being $t \times b$, and if the resultant stress on this

cross-section is a force P which acts at point shown, being eccentric an amount x, then the compression is P and the bending moment is Px. The maximum unit fiber stress will be $\frac{P}{bt} + \frac{Px}{z}$, where z is the section modulus. As $z = \frac{bt^2}{6}$, the maximum unit stress becomes $\frac{P}{bt}\left(1 + \frac{6x}{t}\right)$.

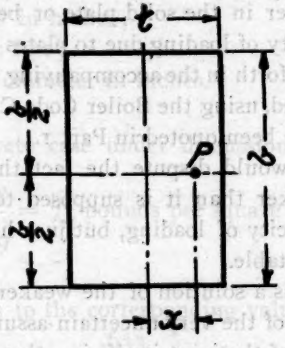


Fig. 1.

Accepting this theory, the problem under discussion lends itself to an accurate analysis *so long as the source of weakness is the plate at the limit of the lapping*. In substantiation of the part in italics is experimental fact, and, in addition, that the efficiency of the plate at this point is decidedly lower than is believed to exist in the joint proper. By "efficiency" of the plate is here meant the ratio of the unit stress, assuming no eccentricity of action to the actual maximum unit stress.

15.—*Scheme of Analysis.*

This discussion does not concern itself with the critical pressure of collapse of a tube subjected to external pressure, but with the detrimental influence of the lap joint.

As this method of analysis cannot be carried out for a general case algebraically except with the result of obtaining

expressions too cumbersome to be useful, the concrete case already taken will be investigated, considering both a single and a double-riveted lap joint.

A cylinder subjected to external pressure is merely two arches placed base to base, one being inverted. Moreover, it is an ideal case for theoretical investigation because, in ordinary arch questions, the distribution of the load is but vaguely known, whilst here it is precisely known. What can be more definitely known or more uniformly distributed than steam pressure?

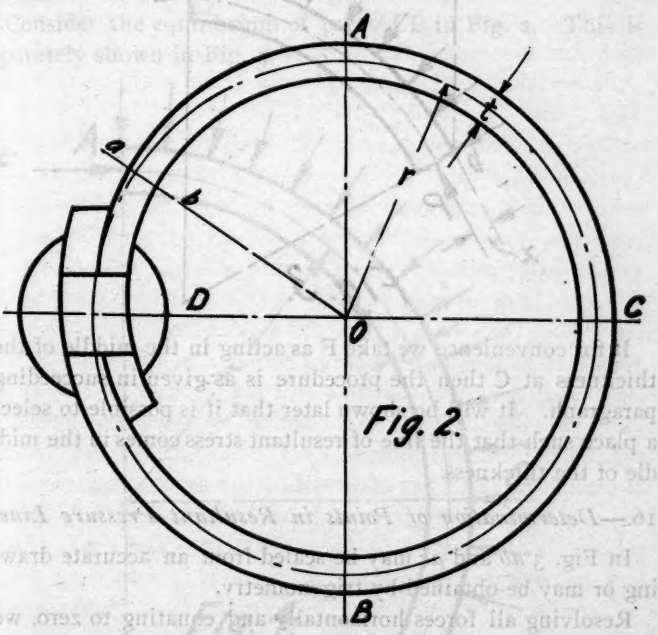


Fig. 2.

In Fig. 2 a section of a single-riveted flue is shown in which the thickness of the plate is exaggerated for convenience. ACB is truly circular and ADB is as nearly circular as possible, *i. e.*, the dot-and-dash line shown is a true circle, constituting the center line of part ACB and cutting the horizontal center line OC between the plates at D. It is required to trace the path of the resultant pressure line around the ring.

Consider the equilibrium of a section of this shell (assume unit thickness perpendicular to paper) such as is shown in Fig. 3. Suppose the resultant stress at the cut section *c* is known in magnitude, direction and position, then the resultant stress at cut section *b* can be found in magnitude, direction and position by the simple laws of static equilibrium, as the forces shown in Fig. 3 constitute a system in equilibrium.

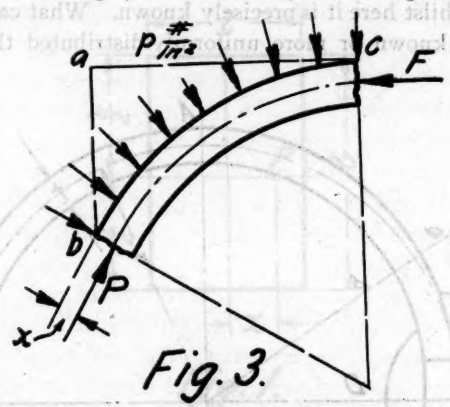


Fig. 3.

If for convenience we take F as acting in the middle of the thickness at C then the procedure is as given in succeeding paragraph. It will be shown later that it is possible to select a place such that the line of resultant stress comes in the middle of the thickness.

16.—*Determination of Points in Resultant Pressure Line.*

In Fig. 3 *ab* and *ac* may be scaled from an accurate drawing or may be obtained by trigonometry.

Resolving all forces horizontally and equating to zero, we have,

$$(\text{Hor. comp. of P}) - F + p.\overline{ab} = 0. \quad . \quad . \quad . \quad (1)$$

Similarly resolving vertically we have,

$$(\text{Ver. comp. of P}) - p.\overline{ac} = 0 \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\therefore \quad P^2 = (\text{Ver. comp. of P})^2 + (\text{Hor. comp. of P})^2$$

$$= (F - p.\overline{ab})^2 + (p.\overline{ac})^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

which obtains P numerically for any given section.

To obtain location of P (letting x be its perpendicular distance from corner b) take moments about b, from which

$$Px = p\frac{\overline{ab}^2}{2} + p.\frac{\overline{ac}^2}{2} - F\left(ab - \frac{t}{2}\right) \quad . \quad . \quad . \quad (4)$$

from which x may be obtained.

Hence if the resultant force at any section be known in magnitude and location the resultant at various other sections may be determined and the locus constructed.

17.—*Path of Resultant Pressure Line.*

Consider the equilibrium of part ACB in Fig. 2. This is separately shown in Fig. 4.

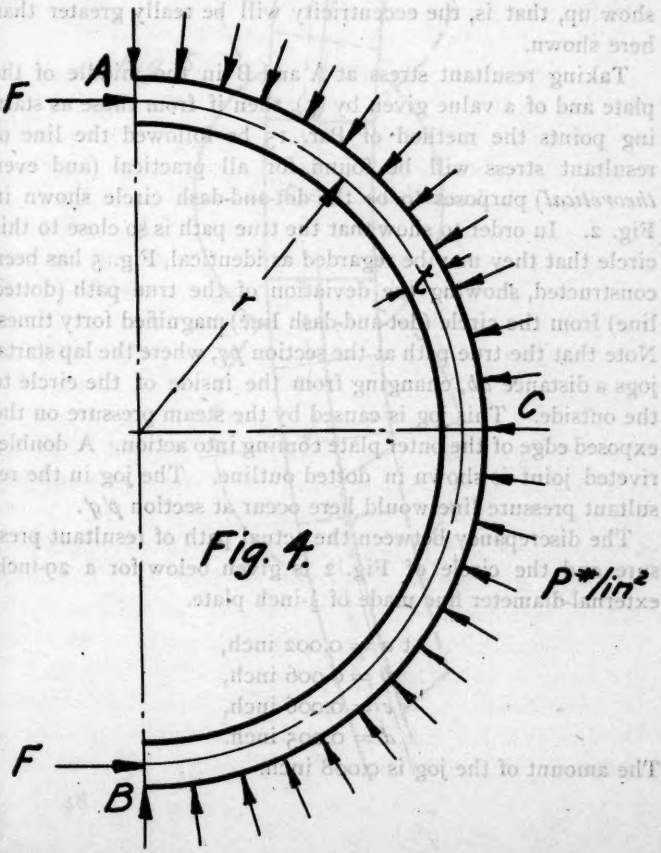


Fig. 4.

If the two forces marked "F," representing the compression in the shell at points A and B, be in the middle of the thickness and satisfy the relation

$$2F = 2p\,(r + t) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

then equilibrium is satisfied. Equilibrium would exist, however, for any position of forces F so long as they were symmetrically situated with respect to OC. A little reflection will satisfy one that they will be in the middle of the thickness if there is no initial bending in the shell. This point need not be further labored, for if F is *not* in the middle of the thickness at A and B so much the worse will the effect of the lapping show up, that is, the eccentricity will be really greater than here shown.

Taking resultant stress at A and B in the middle of the plate and of a value given by (5), then if from those as starting points the method of Par. 15 be followed the line of resultant stress will be found for all practical (and even *theoretical*) purposes to be the dot-and-dash circle shown in Fig. 2. In order to show that the true path is so close to this circle that they may be regarded as identical, Fig. 5 has been constructed, showing the deviation of the true path (dotted line) from the circle (dot-and-dash line) magnified forty times. Note that the true path at the section *pq*, where the lap starts, jogs a distance *ab*, changing from the inside of the circle to the outside. This jog is caused by the steam pressure on the exposed edge of the outer plate coming into action. A double-riveted joint is shown in dotted outline. The jog in the resultant pressure line would here occur at section $p'q'$.

The discrepancy between the actual path of resultant pressure and the circle of Fig. 2 is given below for a 29-inch external-diameter flue made of ½-inch plate.

At $a = 0.002$ inch,
$b = 0.006$ inch,
$c = 0.006$ inch,
$d = 0.005$ inch.

The amount of the jog is 0.008 inch.

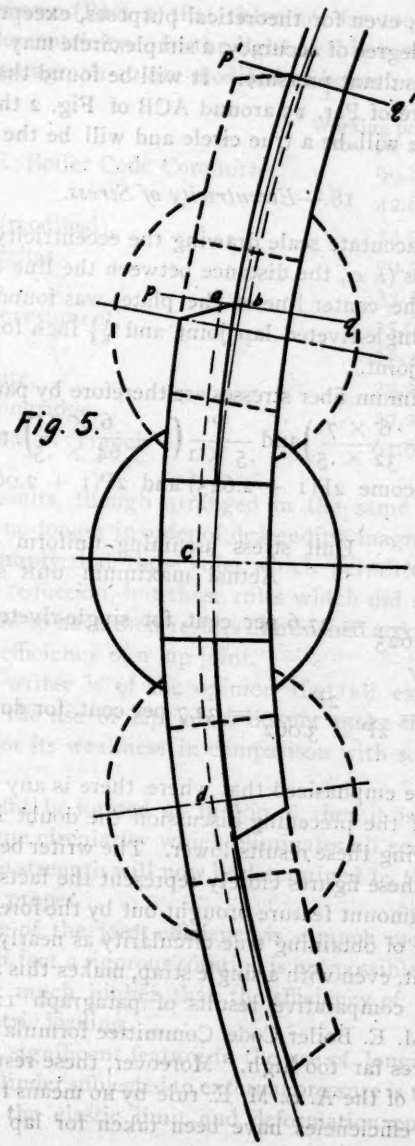


Fig. 5.

Evidently, even for theoretical purposes, except they be of an unusual degree of accuracy, a simple circle may be assumed as line of resultant pressure. It will be found that applying the procedure of Par. 15 around ACB of Fig. 2 the resultant pressure line will be a true circle and will be the center line of the plate.

18.—*Eccentricity of Stress.*

From an accurate scale drawing the eccentricity of the resultant stress (*i. e.*, the distance between the line of resultant stress and the center line of the plate) was found to be $\frac{7}{32}$-inch for a single-riveted lap joint and $\frac{11}{64}$ inch for a double-riveted lap joint.

The maximum fiber stresses are therefore by paragraph 14, $\frac{P}{.5 \times 1}\left(1 + \frac{6 \times 7}{32 \times .5}\right)$ and $\frac{P}{.5 \times 1}\left(1 + \frac{6 \times 11}{64 \times .5}\right)$, respectively.

These become $2P(1 + 2.625)$ and $2P(1 + 2.062)$, respectively.

$$\text{Efficiency} = \frac{\text{Unit stress assuming uniform distribution}}{\text{Actual maximum unit stress}}$$

$$= \frac{2P}{2P \times 3.625} = 27.6 \text{ per cent. for single-riveted lap joint.}$$

Similarly

$$\text{Efficiency} = \frac{2P}{2P \times 3.062} = 32.7 \text{ per cent. for double-riveted lap joint.}$$

It is to be emphasized that where there is any doubt as to the rigor of the preceding discussion the doubt is such that it would bring these results lower. The writer believes, however, that these figures closely represent the facts.

The paramount feature brought out by the foregoing is the importance of obtaining true circularity as nearly as possible. A butt joint, even with a single strap, makes this a possibility.

19. The comparative results of paragraph 12 show that the A. S. M. E. Boiler Code Committee formula gives working pressures far too high. Moreover, these results show up the danger of the A. S. M. E. rule by no means fully, because customary efficiencies have been taken for lap joint, and it

has been shown (Part 2) that these are far too high due to neglected eccentricity. In the light of this new analysis the results of paragraph 12 are modified as below:

	Working pressure, lbs. per sq. in.	Per-centage.
A. S. M. E. Boiler Code Committee,	99.8	100.0
Fairbairn,	42.8	42.9
Fairbairn (modified),	34.7	34.8
Bureau Veritas,	79.3	79 5
Hutton,	33.1	33.2
German Government,	75.6	75.8
Lloyd's,	27.2	27.3
U. S. Statute,	27.2	27.3
Michael Longridge,	27.2	27.3
British Board of Trade,	61.6	61.7

These results, though arranged in the same sequence as before, are no longer in order of descending magnitude. The reason is simply that those rules which permitted lap joints suffered no reduction, but those rules which did not permit a lap joint had to have their results diminished according to the new lower efficiency of a lap joint.

20. The writer is of the opinion that all existing rules permitting the use of lap joints do not make the necessary allowance for its weakness in comparison with solid plate or butt joints.

If the joint be formed as in Fig. 6, then it is possible to maintain true circularity which eliminates all eccentricity of action. The strength will now be determined by the efficiency of the joint proper.

The value of the joint efficiency is a much more complex problem—in fact a rigorous solution is not possible; it would, however, be much higher than the efficiency of the plate is under eccentric loading.

The most significant feature in the use of longitudinal lap joints for cylinder subjected to external pressure is that, should stress reach the elastic limit and deformation ensue, the *ec-*

centricity of action is increased and collapse inevitable. In cylinders subjected to internal pressure, on the other hand, deformation decreases eccentricity and restores satisfactory conditions.

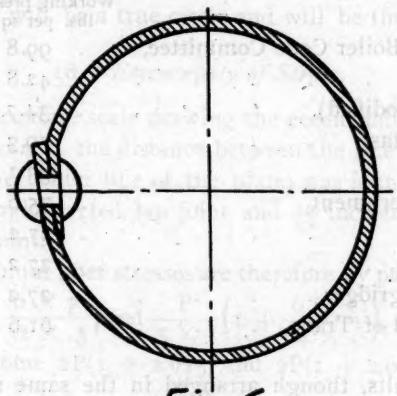


Fig. 6.

Considering that the use of longitudinal lap joints is rapidly diminishing in good practice for internal pressure, is it not desirable to prohibit absolutely the use for external pressure; or, what would serve equally well, cut down the acknowledged efficiency to its true value. If this were done the use of a lap joint would entail such an abnormally thick plate to withstand a given pressure that considerations of economy would probably cause its use to cease.

Appendix.

To find the maximum efficiency of a single-riveted lap joint considered in the conventional manner.

Let p = pitch of rivets;
d = diameter of rivets;
t = thickness of plate;
p_b = unit bearing strength;
f_s = unit shear strength;
f_t = unit tensile strength.

Equating failure by shearing, tearing and crushing, we get

$$(p - d)\, tf_t = \frac{\pi d^2}{4} f_s;$$

$$(p - d)\, tf_t = dtp_b;$$

or

$$d = \frac{p_b}{f_s} \cdot \frac{4}{\pi} \cdot t = \frac{95{,}000}{42{,}000} \cdot \frac{4}{\pi} t = 2.88t.$$

The strengths of material used are those given by the Boiler Code Committee of the A. S. M. E. for M. S. These are the same as given by the Massachusetts Steam Boiler Rules:

$$p = d + \frac{p_b}{f_t} d = \left(1 + \frac{95{,}000}{55{,}000}\right) d = 2.727d.$$

$$\text{Efficiency} = \frac{p - d}{p} = \frac{1.727d}{2.727d} = 63.3 \text{ per cent.}$$

To Find the Maximum Efficiency of a Double-Riveted Lap Joint.

With notation as before, equating failure by shearing, tearing and crushing we get

$$(p - d)\, tf_t = 2\, \frac{d^2}{4} f_s;$$

$$(p - d)\, tf_t = 2dtp_b;$$

or

$$d = \frac{p_b}{f_s} \cdot \frac{4}{\pi} \cdot t = 2.88t;$$

$$p = d + \frac{2p_b}{f_t} d = 4.45d.$$

$$\text{Efficiency} = \frac{p - d}{p} = \frac{3.45d}{4.45d} = 77.6 \text{ per cent.}$$

NOTES.

MARINE STEAM BOILERS AND BOILER ROOM EQUIPMENT.*

By Chas. F. Bailey, Mem. S. N. A. & M. E.

Chief Engr. Newport News Shipbuilding and Dry Dock Co., Newport News, Va., U. S. A.

During the last decade there has been a considerable development and improvement in marine boilers, both of the Scotch type and of the water-tube type. The changes, however, in design and improvement in efficiency have been less marked than the great changes in marine propelling machinery, in which the steam turbine has largely replaced the reciprocating engine in naval work and in much of the merchant work. This condition is now being modified by the fitting of speed-reduction devices. In submarine boats, a number of other naval vessels and in some merchant vessels, the crude-oil engine is being adopted.

By comparison, therefore, the developments in marine boilers appear to be overshadowed, as no radically new principles have been adopted in practice, although several notably conspicuous new inventions have been made which may develop in the future. Among these should be mentioned the Bone surface-combustion experiments, the mercury boiler, invented by Mr. W. L. R. Emmet, the Talbot high-pressure boiler, which has been used in small units with much success, and the Bettington powdered-coal boiler.

The efficiency of marine boilers has been generally improved due to the following causes: A more careful study of conditions governing combustion in the boiler and means for quick analysis of the products of combustion and for observing the variation of temperatures throughout the path of the gases; the study of heat transfer, in which is involved the volume of the combustion chamber, baffling of the gases, and arrangement of passages for same, tube spacing, disposition of the heating surfaces, circulation of the water, liberation of the steam, cleaning of the heating surfaces, both on the fire side and on the water side. Steam pressures cannot be said to have generally increased excepting in some particular instances.

In addition to the foregoing there should be mentioned the revival of interest in superheated steam, which bids fair to largely increase and which promises considerable gain in overall efficiency. Oil burning has developed to a high degree of perfection both with steam and air atomization, and notably with mechanical spraying. The study of corrosion has occupied considerable attention.

Improved workmanship is recognized as an important factor in ultimate boiler efficiency, and along with this may be noted the advance in shop facilities for working the materials and tools for more accurately and rapidly machining the finished parts. This has led to the standardizing of many details which have been most carefully designed for assembly, or replacement, durability, reduction in weight, improvement in appearance and facility of manipulation.

*A paper presented at a meeting of the International Engineering Congress, 1915, in San Francisco, Cal., September 20-25, 1915.

The use of the oxy-acetylene flame for cutting materials, the use of electric welding, especially for repairs, and the extensive adoption of pneumatic tools have had a material effect in improving the quality of boiler work and reducing the cost of construction and upkeep.

Advances have been made in mechanical draft, feed purification and supply in the handling and disposal of refuse and ashes; fireroom communication; and time-indicator firing has been largely adopted in the highest-grade work. These various minor improvements have resulted in a considerable gain in efficiency and reduced cost of upkeep.

The increase in size of marine boiler installations is one of the most striking recent advances, and along with this increase is to be noted some decrease in heating surface per horsepower, which is largely due to reduced steam consumption of the propelling machinery and partly due to a higher boiler efficiency and dependability. In naval work the use of oil fuel only, or oil and coal combined, has reduced the heating surface to a more marked degree. Boiler installations of over 100,000 horsepower per ship are now made, whereas ten years ago powers of 20,000 to 30,000 horsepower per ship were about the maximum.

SCOTCH BOILERS.

The changes during the past decade in the Scotch type of boiler and other fire-tube boilers have been less than in most forms of water-tube boilers.

We note simplification in design due to the modern rolling-mill facilities for supplying larger plates, thus reducing the number of seams. It is usual now to construct single-end Scotch boilers up to 12 feet (3,660 mm.) in length and nearly 18 feet (5,486 mm.) in diameter with no middle circumferential seam and with only one seam across the back head. Separate combustion chambers are usually fitted in single-end boilers and also in double-end boilers in connection with forced draft. With natural draft or assisted draft, in double-end boilers, it is not unusual to connect the corresponding furnaces in opposite ends of the boiler to straight through combustion chambers; this considerably reduces the length and weight and simplifies the construction. When such combustion chambers are fitted, fire-brick walls are sometimes built up in the middle between the furnaces.

The combustion chambers for double-end boilers are usually tied to the bottom of the boilers to prevent working when steaming, although with single-end boilers it is not essential to tie the chambers down.

The lower wrapper plates of the combustion chambers are sometimes made in one plate of greater thickness at the bottom part where unsupported by stays, thus saving the side-riveted laps.

The question of material of boiler tubes is one on which there is a considerable difference of opinion. The best charcoal-iron tubes and seamless-drawn steel tubes seem under favorable conditions to give equally satisfactory results. The use of Serve tubes in Scotch boilers has practically been discontinued. With water-tube boilers it is common to use seamless-drawn steel tubes.

It is now usual to fit the back ends of the corrugated furnaces either of the horse-collar type, the straight type or of some form of flanged type which permits of the furnaces being renewed at a minimum expense.

It is not uncommon to fit Scotch boilers of very large dimensions. The boilers of the S. S. *Aquitania* are 17 feet 8 inches (5,385 mm.) mean diameter by 22 feet 0 inches (6,706 mm.) mean length.

Some owners take precautions to prevent air drafts from circulating at the bottoms of their boilers, and lag the bottoms with asbestos or magnesia blankets or pads which are easily removed for examination. This is an excellent practice, promoting economy in operation and upkeep.

The ordinary plate saddle or support is largely used for Scotch boilers in conjunction with fore-and-aft chocks. A form of Scotch boiler support

which is frequently adopted by American builders is illustrated in Fig. 1. These supports consist of heavy bent-steel lug plates riveted to the lower shell of the boiler, each lug being fitted with a ¼ inch (6.3 mm.) caulking plate adjacent to the shell. These plates can be hydraulically riveted when the shell riveting is being done. Girders are fitted in the ship, as indicated in Fig. 1, to which the lug plates are bolted. Provision is made at one end of the boiler for expansion. Similar supports are also common for Yarrow boilers. All boiler supports should provide the greatest facility possible for care of the lower parts of the boiler. It may be noted that the heating surface per cubic foot volume of boilers has not altered materially, but the tendency is to allow more steam space, which gives better access for cleaning and care.

WATER-TUBE BOILERS.

The recent developments in the design and construction of marine water-tube boilers have been much more pronounced than in Scotch boilers. Many of the early errors in design have been eliminated, so that high efficiencies are now common, both in naval and merchant work.

The requirements demanding the best material and workmanship have been recognized and now many of the water-tube boilers are examples of the finest product of skilled designers and workmen and modern machinery and methods. In this connection it is interesting to refer to the early influence of Mr. A. F. Yarrow, who, in 1891,* pointed out that only the best materials and workmanship should be employed in boiler work.

Boilers of the Babcock & Wilcox, Niclausse, Belleville and Yarrow types have all retained their essential features for the last ten or fifteen years. Many of the other so-called express types of boilers have been merging towards a more or less composite design in which the objectionable features have been largely eliminated.

Boilers which a few years ago were fitted with tubes having extreme bends are now made with these tubes of gradual bends, and in some cases, the tubes are straight with the exception of the lower ends entering the water drums. The upper ends of the tubes are now usually submerged and arranged to facilitate cleaning and inspection; air pockets are generally avoided as it has been found corrosion is active at these points. The straight-tube type of boiler, represented by the Yarrow, fits the rows of tubes next to the fire with slight bends to allow for expansion. Steam drums are of larger diameter in the three-drum or express type of boiler than in the Babcock & Wilcox or the Niclausse boiler, due to structural reasons, accessibility, and to provide a good volume of water.

The water pockets at the lower ends of the tubes are approaching the circular section and in some cases the oval or D-drum has been replaced by the circular drum.

Due to these improvements, resulting in great reliability and increased durability, the heating surface for naval work, especially where oil fuel is used, has been notably reduced.

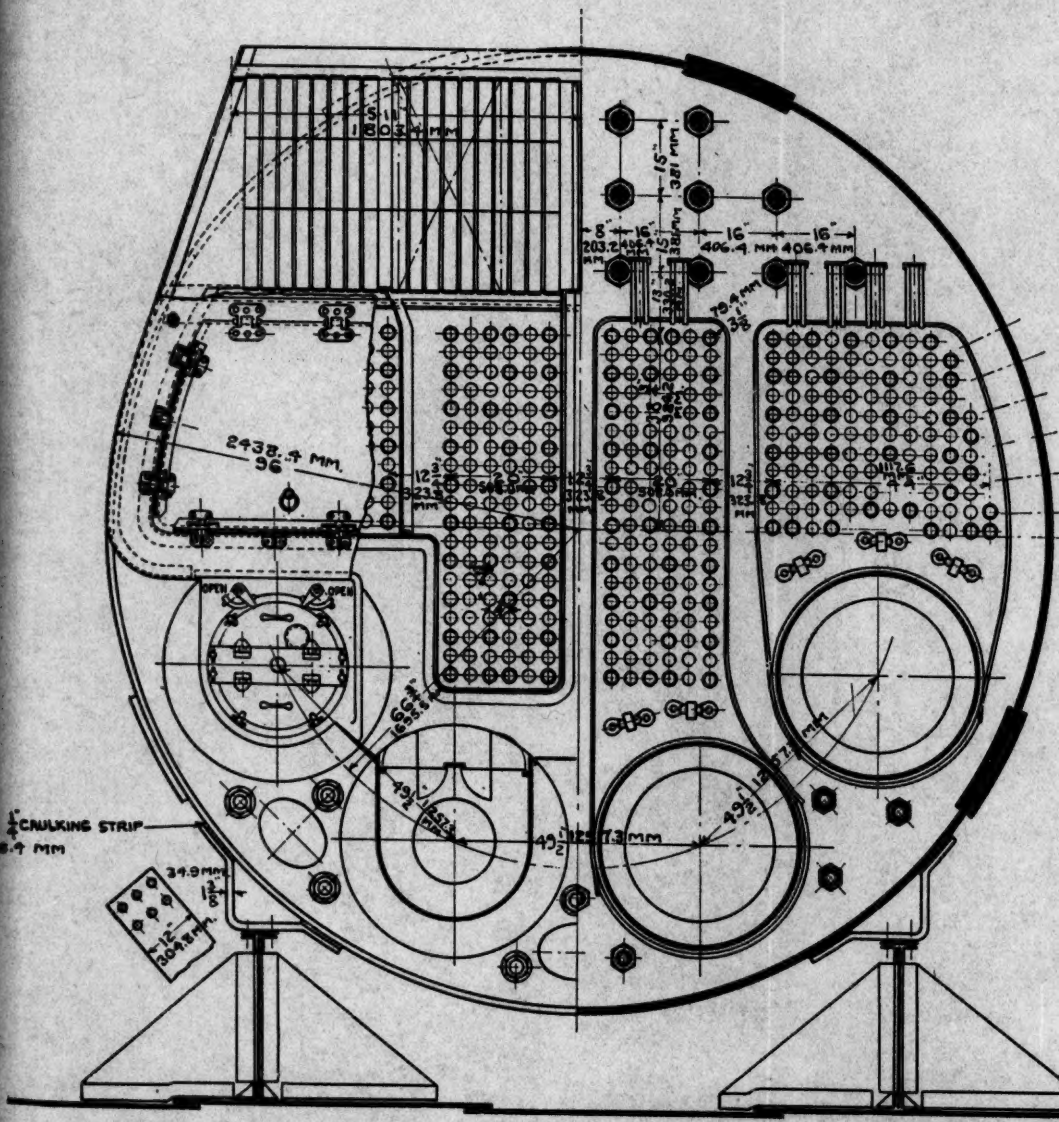
The water-tube boiler has been adopted for practically all naval vessels, excepting auxiliaries such as colliers, where Scotch boilers are used in some cases.

The German Navy uses the Schulz or Schulz-Thornycroft boiler nearly exclusively for all sizes of vessels. Other navies are using two or more types of boilers, principally as follows:

England—Babcock & Wilcox and Yarrow for large vessels. Yarrow and White Forster for smaller vessels and destroyers.

France—Belleville, Niclausse and Guyot Du Temple for large vessels; Normand and Lagrafel d'Allest, Du Temple Guyot and White Forster for small vessels and destroyers.

* Paper on Construction of Boilers Adapted to Forced Draught, "Institution of Naval Architects," 1891.


CAULKING STRIP
6.4 MM
OPEN
OPEN
2438.4 MM.
96"
406.4 MM
406.4 MM
203.2 MM
8"
16"
16"
16"
15"
381 MM
304.8 MM
12"
304.8 MM
323.8 MM
12 1/2"
323.8 MM
49 1/2"
1257.3 MM
49 1/2"
1257.3 MM
49 1/2"
1257.3 MM
24.9 MM
1"
12"
304.8 MM.

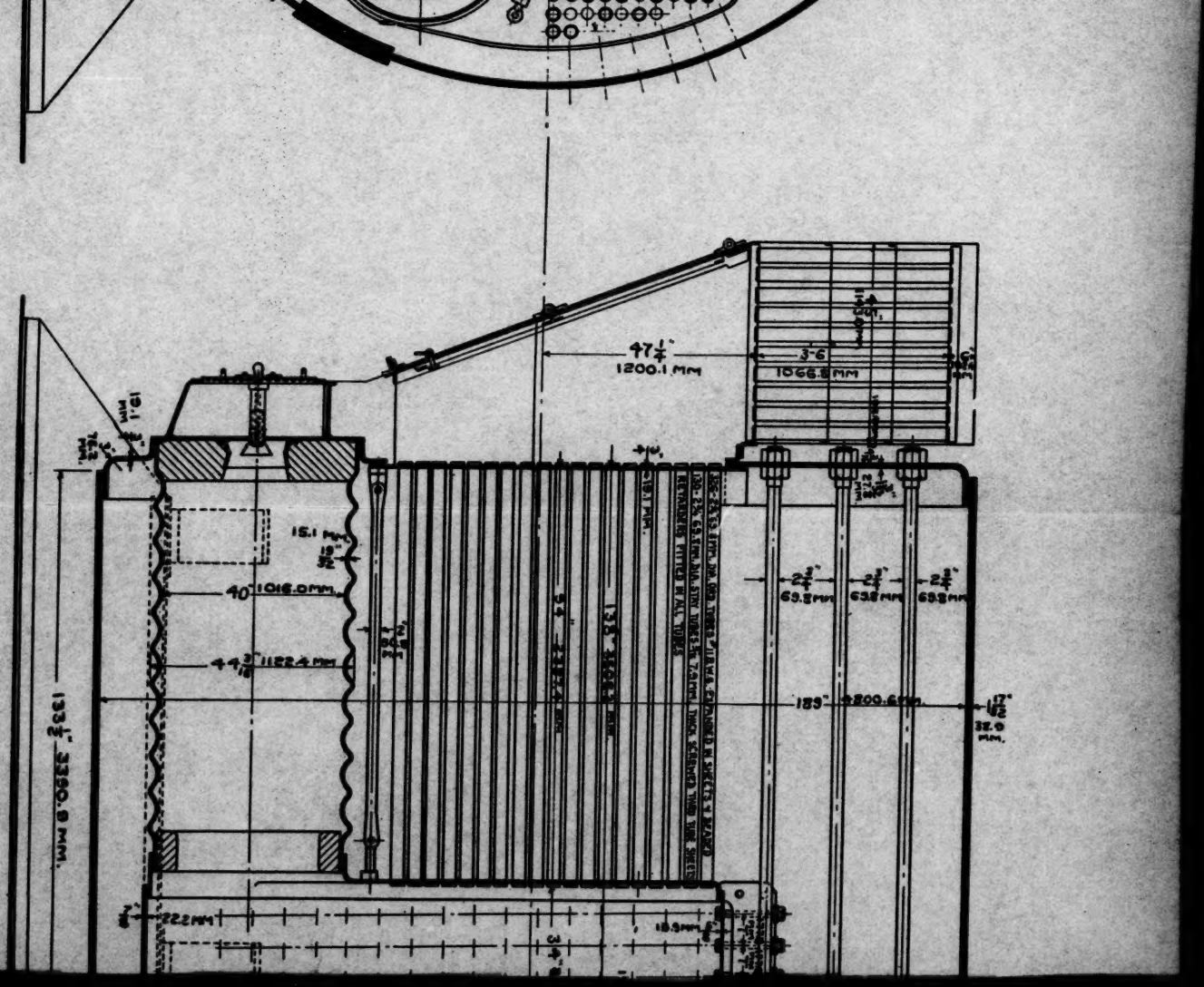

47 1/4"
1200.1 MM
3'-6
1066.8 MM
40
1016.0 MM.
44 3/16"
1122.4 MM
133 1/2"
3390.9 MM.
54"
1371.6 MM.
135"
3429.0 MM.
189"
4800.6 MM.
2 3/4"
69.8 MM
2 3/4"
69.8 MM
2 3/4"
69.8 MM
15.1 MM.
19.1 MM.
222 MM.
18.9 MM.

FIG. 1.

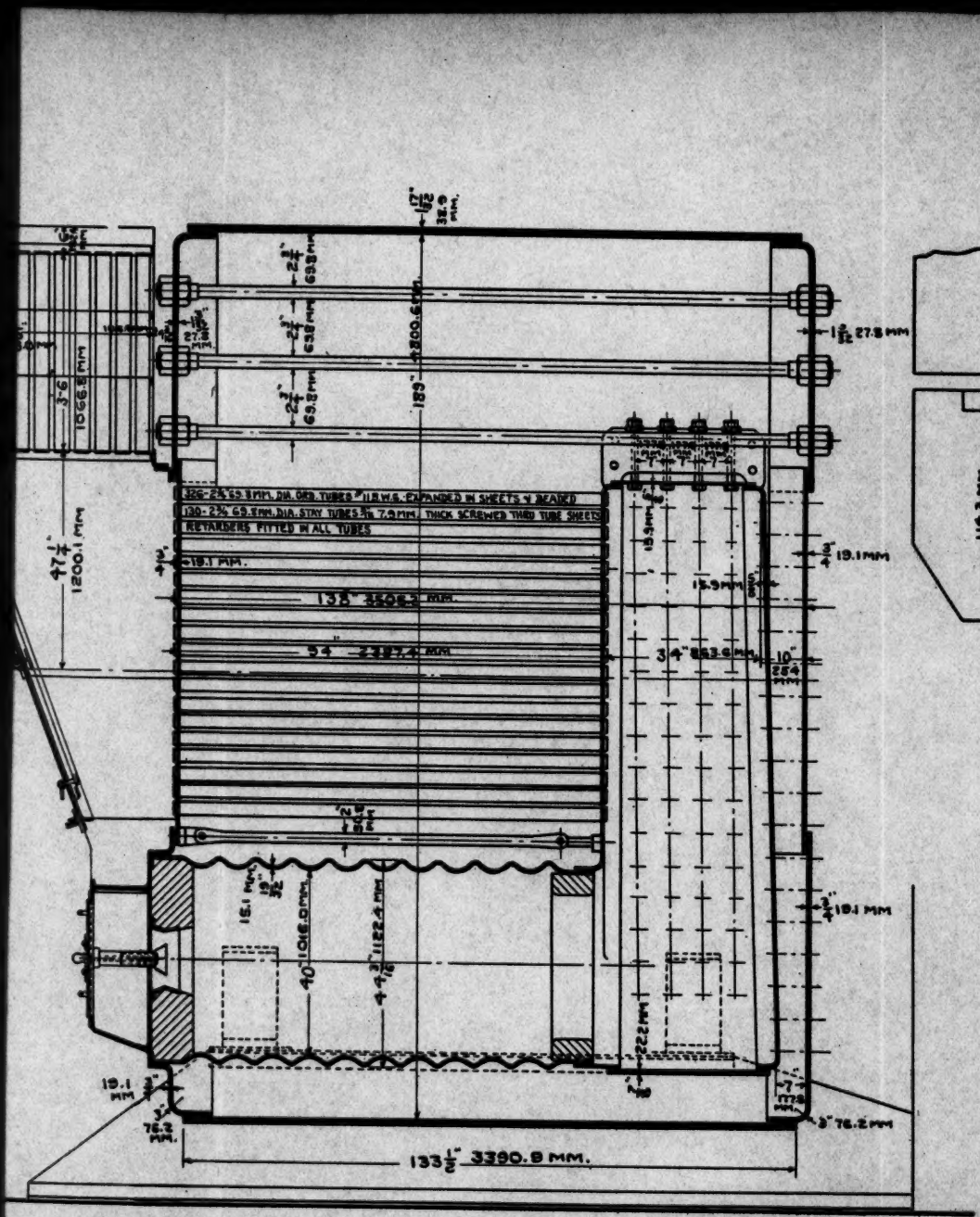
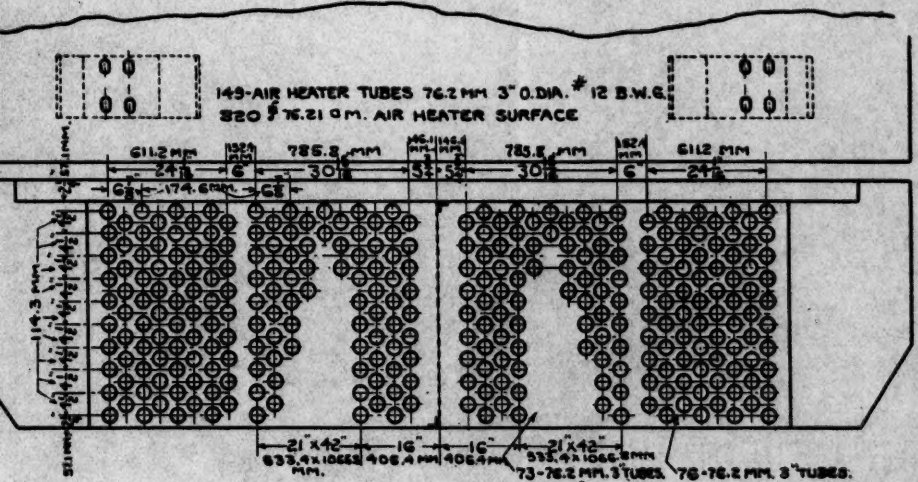

326-2¼" 53.8 MM. DIA. OFD. TUBES P. I.I.R.W.S. EXPANDED IN SHEETS & BEADED
130-2¼" 69.8 MM. DIA. STAY TUBES ⅝" 7.9 MM. THICK SCREWED THRU TUBE SHEETS
RETAINERS FITTED IN ALL TUBES
138" 3505.2 MM.
34" 2387.6 MM.
34" 863.6 MM.
133½" 3390.9 MM.
SC
FIT

FIG. 1.



DATA	1 BOILER
EXTERNAL DIA. TUBES	2¾" 69.8 MM.
NUMBER OF TUBES	456
LENGTH BTWN T.S.	92½" 2349.3 MM.
SURFACE OF TUBES	2530 ft² 235.15 M².
SURFACE OF CC	358 ft² 33.85 M².
SURFACE OF FURNACE	214 ft² 19.89 M².
TOTAL HEATING SURFACE	3100 ft² 288.1 M².
NET AREA THRU TUBES	14.4 ft² 1.338 M².
STEAM PRESSURE	200 LBS. H.W. 180 PER SQ.IN.
WATER TEST PRESSURE	300 LBS. 21.09 KGS. PER SQ.CM.
HEIGHT ABOVE CC & LS. DIA	275

SCOTCH BOILER AND AIR HEATER BOX

FITTED FOR BURNING FUEL OIL WITH FORCED DRAFT

S.S "TOPILA"

Italy—Belleville, Niclausse, Blechynden, Babcock & Wilcox and Yarrow for large vessels; Thornycroft for small vessels.

Japan—Yarrow, Belleville, Niclausse, Miyabara and Babcock & Wilcox for large vessels. Miyabara and Yarrow for small vessels.

United States—Babcock & Wilcox and Yarrow for large vessels. Babcock & Wilcox, Yarrow, Normand, Thornycroft and White Forster for intermediate vessels and destroyers.

Thus it is seen that a wide range in type of boiler is still being allowed by naval authorities. The requirements in boiler installation which appeal to one nation as most essential appear differently to another, and doubtless properly so, as the broadest view of ultimate efficiency should be taken, into which enter a consideration of the home facilities for building, replacing or repairing in the most expeditious manner. Thus the fine shops in Great Britain and the United States for expeditiously building the Babcock & Wilcox boiler and the corresponding facilities in France for constructing the Niclausse boiler and the absence of such special works in Japan, may influence the Japanese Admiralty in adopting in a home-built vessel the Miyabara boiler or some other type which may be constructed with the shop facilities at hand.

It is to be noted that the sizes of water-tube boiler units have greatly increased. The efficiency of the large units is higher than that of the small units, especially with oil firing.

In installations of the small-tube boilers for large powers it is usual, in English and American practice, to provide considerable more heating surface than with installations of Babcock & Wilcox boilers. In making comparisons between boilers on the basis of heating surface this point should be borne in mind.

WATER-TUBE BOILERS IN MERCHANT VESSELS.

In considering the installation of water-tube boilers in trans-ocean or coasting merchant vessels, the question of proper facilities for cleaning the boilers during long voyages must be kept in mind. Many boilers which show high efficiency under shop tests may fail under the severe conditions of service in merchant shipping, where long voyages are made, changing weather conditions are encountered and the quality of coal is sometimes inferior. The operation of merchant vessels is often attended with limited time in port for overhauling, repairs and cleaning. Some large installations of water-tube boilers have been made in merchant work which have not given the satisfactory service anticipated, owing to the lack of facilities for properly cleaning the fires, sweeping the heating surfaces, regulating the air to the furnaces, and ventilating the stokeholds. Such installations do much to retard the introduction of water-tube boilers in the merchant marine. The Babcock & Wilcox boiler has probably been installed in a greater number of mercantile vessels than any other type of water-tube boiler, and its success has been largely due to the characteristics outlined by the late Rear-Admiral George W. Melville,* formerly Engineer-in-Chief, U. S. Navy, in an article in which he says:

"From my study of the subject, I had reached the conclusion that the thoroughly satisfactory water-tube boiler should possess, among others, the following characteristics:

"Reasonable lightness, with scantlings sufficient to promise reasonable longevity;

"An adequate amount of water, so that failure of the feed supply or any inattention thereto would not immediately cause trouble;

"Accessibility for cleaning and repairs on both water and fire sides;

"Straight tubes, with no screw joints in the fire, but the simple expanded joints so well tested out for years;

* "The Development of the Marine Boiler in the Last Quarter Century," "Engineering Magazine," January, 1912.

"No cast metal, either iron or steel, subjected to pressure;
"Ability to raise steam quickly;
"High economy of evaporation;
"Economy of space;
"Interchangeability of parts, and, as far as possible, the use of regular commercial sizes so that repair material could be procured anywhere;
"The ability to stand severe forcing without injury;
"The ability to stand abuse—that is, to be of rugged construction and not so delicate as to require skilled mechanics to run it;
"Safety against disastrous explosion, meaning that only the part of the boiler which gave way would be damaged."

Higher efficiencies can often be obtained with well-designed water-tube boilers than with Scotch boilers, principally owing to the fact that the furnace is large, its form may be nearly ideal for thorough combustion before the gases enter the tubes; whereas in Scotch boilers the furnaces are contracted and low and the combustion space is completely surrounded by the cooler boiler surfaces.

Experience in the merchant vessels *Creole, Matsonia, Adeline Smith,* and several others, each operated by different owners and fitted with Babcock & Wilcox boilers, indicates that with a reasonably intelligent engineering staff, boilers of this type are thoroughly reliable and can be operated at a lower cost of maintenance and are more flexible, and probably more efficient, than ordinary Scotch boilers. In the *Adeline Smith,* fitted with four boilers, there has never been a delay of any kind for cleaning or minor repairs; one of these boilers can be cooled sufficiently to work in within an hour or two, and steam raised again in forty-five minutes.

The writer is informed in regard to the six Babcock & Wilcox boilers and three single-end Scotch boilers installed in the steamer *Matsonia,* and using oil fuel, that very little cleaning of the water-tube boilers is necessary and that it is not required to mechanically clean the tubes oftener than once in two or three months. No signs of corrosion have appeared. The zinc plates are renewed approximately each six months. A half keg of Renown compound is also used per trip of 2,000 miles. It is not found necessary to have higher-grade men in the fireroom to operate these boilers than those ordinarily operating Scotch boilers. Feed-water regulators are desirable and the Hough patent feed-check valve is considered a necessity where reciprocating feed pumps are used.

The Matson Navigation Company's purpose in installing a combination of Scotch boilers and Babcock & Wilcox boilers in this vessel was that they might have Scotch boilers for winch work in port, owing to the larger steam space. They find, however, that a number of steam schooners fitted with only two Babcock & Wilcox boilers have no boiler troubles even though using them constantly in port for donkey boiler service.

The saving in weight and space which may be effected with water-tube boilers over that required for Scotch boilers is great, as shown by Admiral Melville in his paper.

The Niclausse Boiler.—This well-known boiler, which has been largely adopted by the French Navy and others, has retained the essential principles of its unique construction, but recent improvements have resulted in a very high efficiency.

These improvements, some of which have been in use a considerable time, include the following:

Constructing the headers of solid-drawn pressed steel of larger rectangular section.

Making the tubes solid-drawn with swellings to form the cones and lanterns.

The adoption of a system of forced circulation and feed distribution to the lower tubes which are nearest the furnace, which also includes arrangements for purifying the feed.

All of these points are described in a paper by M. Niclausse* and records are given of a test by naval officers of the boiler as proposed for the new French Battleship *Béarn*.

Figures 2, 3 and 4 indicate the arrangement and some details of this high-duty Niclausse boiler.

White Forster Boiler.—As is well known, the distinctive features of this boiler are the adoption of a uniform curvature of all of the tubes in any particular boiler and the arrangement of the tubes in the steam and water drums so as to permit any particular tube to be removed and replaced through the steam drum without disturbing other tubes in the boiler

The spare tubes may be carried of the longer lengths and cut to any desired length.

This boiler is now constructed either with D-shaped water drums or circular water drums. Fig. 5 shows the latest type of the White Forster Boiler as built in the United States by the Babcock & Wilcox Company, which embodies the circular water drums.

Owing to the large radius of curvature of the tubes they are easily cleaned, both internally and externally, and the stresses due to expansion are minimized.

This boiler is capable of developing a very high efficiency and it can also be severely forced without injury.

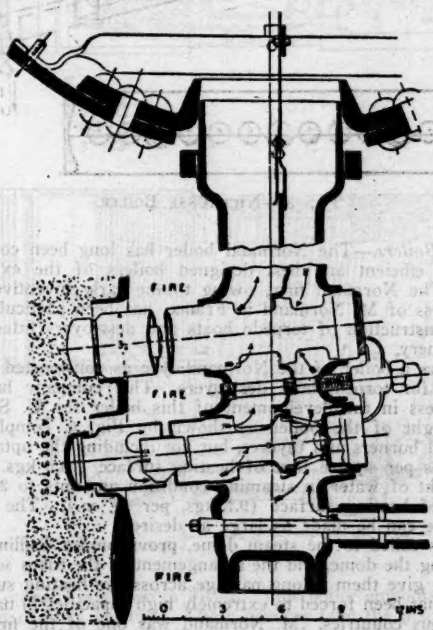


FIG. 2.—NICLAUSSE BOILER.

* Paper read before the Mechanical Engineers at Paris, July 8, 1914, and published in "Engineering," July 17, 1914.

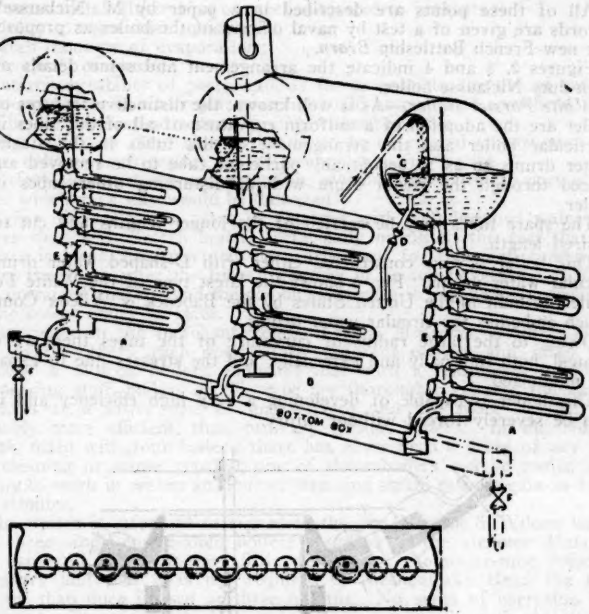


FIG. 3.—NICLAUSSE BOILER.

Normand Boilers.—The Normand boiler has long been considered one of the most efficient and best designed boilers of the express water-tube type. The Normand firm, owing to the early inventive ability and resourcefulness of M. Normand in France, has been particularly successful in the construction of torpedo boats and destroyers fitted with high-power machinery.

Fig. 6 shows a boiler of the Normand type as constructed by the Bath Iron Works for torpedo-boat destroyers. This company has met with singular success in the development of this boiler for U. S. destroyers. The dry weight of the boiler as shown by Fig. 6, complete with all mountings, oil burners and tuyéres, but not including the uptake, amounts to 11.2 pounds per square foot of heating surface (54.6 kgs. per sq. m.) and the weight of water in steaming condition amounts to 2 pounds per square foot of heating surface (9.7 kgs. per sq. m.). The combustion-chamber space can be made as large as desired.

Attention is called to the steam dome, provision for baffling the steam before entering the dome, and the arrangement of the tubes so as to baffle the gases and give them a long passage across the heating surface.

The boiler has been forced to extremely high capacities in naval installations in various countries. M. Normand was one of the first engineers to point out the ill effect of the admittance of feed in such a manner as to interrupt the proper circulation in water-tube boilers and to show the inefficiency resulting from such methods of feeding.

The Box Type of Boiler.—The Box Type of boiler, built by the Babcock

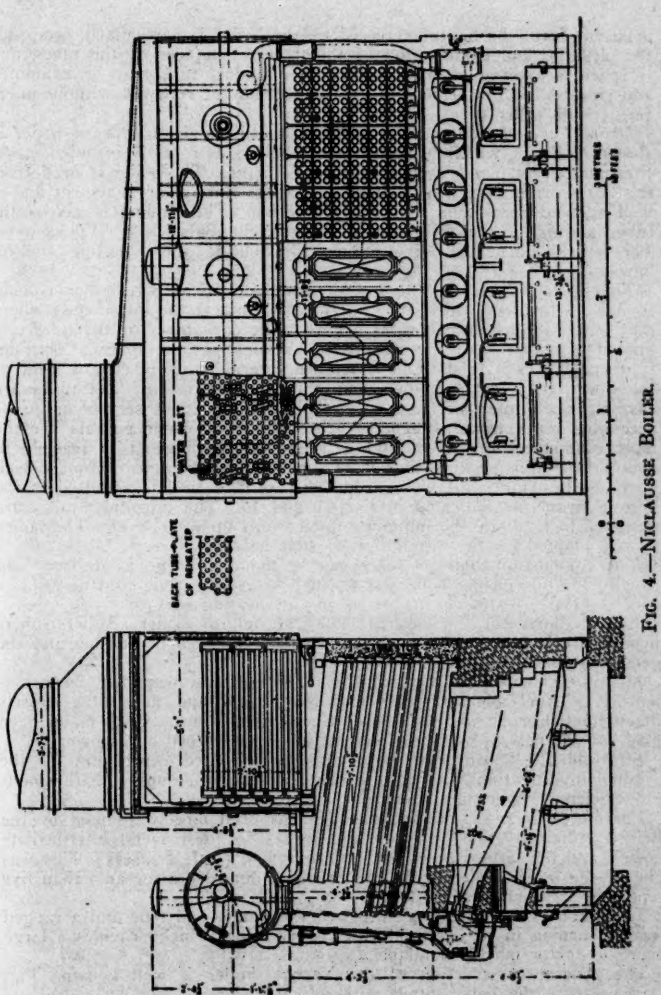


FIG. 4.—NICLAUSSE BOILER.

& Wilcox Company is shown in Fig. 7. This boiler embodies an ingenious arrangement of steel headers. These headers take the place of the usual water drums in the A type of boiler, and it should be noted that they may be of either the straight box or corrugated form, running either longitudinally or crosswise of the bank of tubes; this allows great flexibility in regard to size of boiler. These headers are of the regular type incorporated in the Babcock & Wilcox boiler, as shown in Fig. 16. As is

apparent from the figures referred to, the boiler is particularly accessible for cleaning and examination of the interior surfaces of the tubes, and if space is allowed in the vessel, any individual tube may be examined and cleaned from either end, and any tube may be renewed without interfering with other tubes.

Drum Type of Boiler.—The Babcock & Wilcox drum type of boiler is shown in Fig. 8, which represents these boilers as recently installed in the Steamers *Great Northern* and *Northern Pacific*. The boiler is fired from the water-drum side, which gives an efficient furnace arrangement and is well adapted for either coal or oil burning. The baffling is across the tubes and is of the well-known and efficient Babcock & Wilcox type. The boilers illustrated were fitted with Schütte and Koerting fuel-oil burners.

Thornycroft Boilers.—The latest form of Thornycroft boiler is constructed with tubes which are straight excepting at the lower ends where they enter the cylindrical water drums. The curvatures of the tubes are limited to only two or three radii, which simplifies the spares. This design enables each tube to be examined internally by placing a light at the lower end; it also allows flexibility for expansion. The tubes are easily cleaned internally and are so disposed in cross section as to be accessible for external sweeping. The cylindrical water pockets are the most economical in weight and cost of construction. It is feasible to make the water pockets of welded or solid-drawn construction, which obviates the dangers of leaky seams in case cold feed is admitted to the bottom drums as indicated in Figs. 9 and 10. The downtake tubes are designed to facilitate the minimum fluctuations in water level. The boiler is well adapted for the installation of superheaters.

The combustion-chamber space can be made as large as desired, and usually for oil-burning boilers is about 0.09 to 0.12 cubic foot per square foot (0.0274 to 0.0366 cu. m. per sq. m.) of heating surface.

Fig. 9 shows the Thornycroft latest design of boiler. When superheaters are installed they are arranged between the boiler casing and the generator tubes as shown in Fig. 10.

Messrs. Thornycroft & Company, Limited, have supplied their system of oil-fuel apparatus for numerous naval and mercantile vessels throughout Europe and America. Such installations, which have been made or are now being made, amount to over 2,000,000 horsepower.

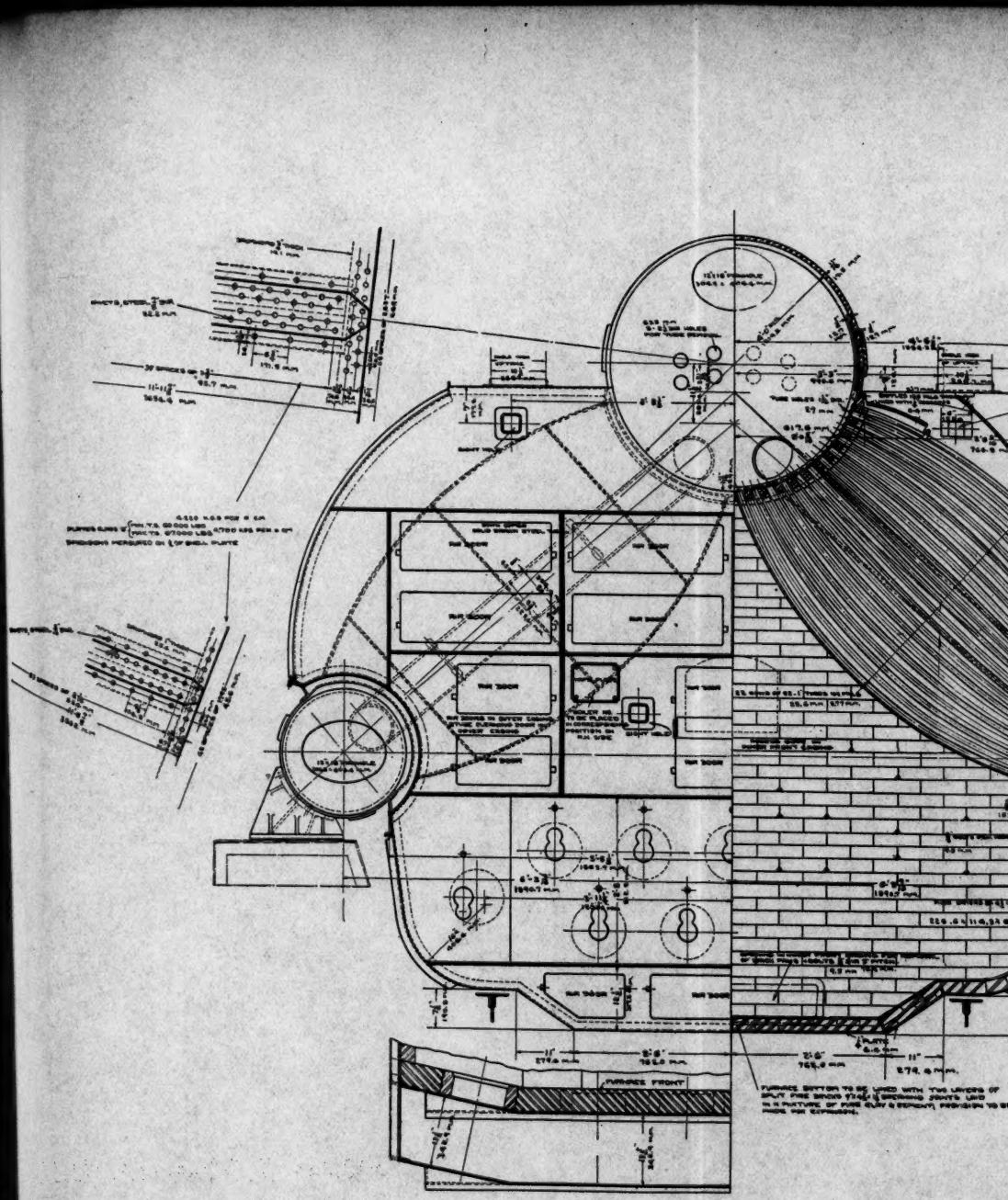
Sir John I. Thornycroft is a particularly noted contributor to the original investigations and developments which are resulting in the modern efficiency water-tube boiler.

Yarrow Boilers.—The Yarrow boiler has been largely adopted by the British Admiralty and by other admiralties for destroyers, intermediate vessels and the largest types of battleships and battle cruisers. This was one of the boilers selected by the British Admiralty after an exhaustive report by its committee in 1902.

These boilers are sometimes fitted of the double-end type and arranged for oil burning in conjunction with coal burning, which provides a large overload factor quickly available.

The general construction of the Yarrow boiler is well known. Fig. 11 represents the boiler fitted without superheaters; Fig. 12 represents the boiler fitted with superheaters; Fig. 13 shows the boiler as now usually fitted with angle-iron baffles and longitudinal feed division plates, as recommended by Messrs. Yarrow.

The high class of work done by the Yarrow Company, both in design and construction, and the large number of experiments which have been conducted in studying the principles of correct boiler construction, theory of operation and efficient working have placed the Yarrow boiler in an enviable position as regards efficiency, endurance, and other important



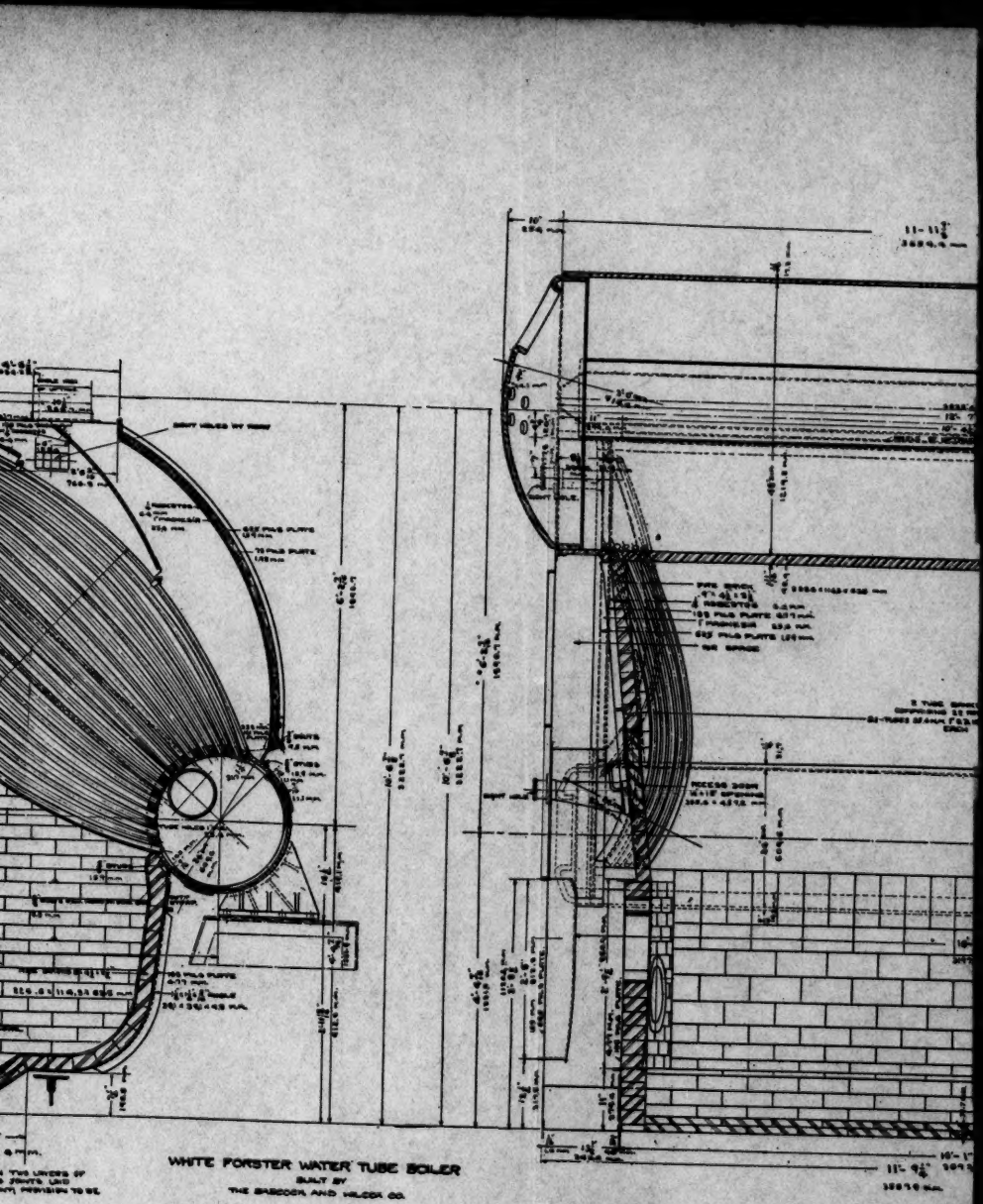

WHITE FORSTER WATER TUBE BOILER
BUILT BY
THE BABCOCK AND WILCOX CO.

FIG. 5.

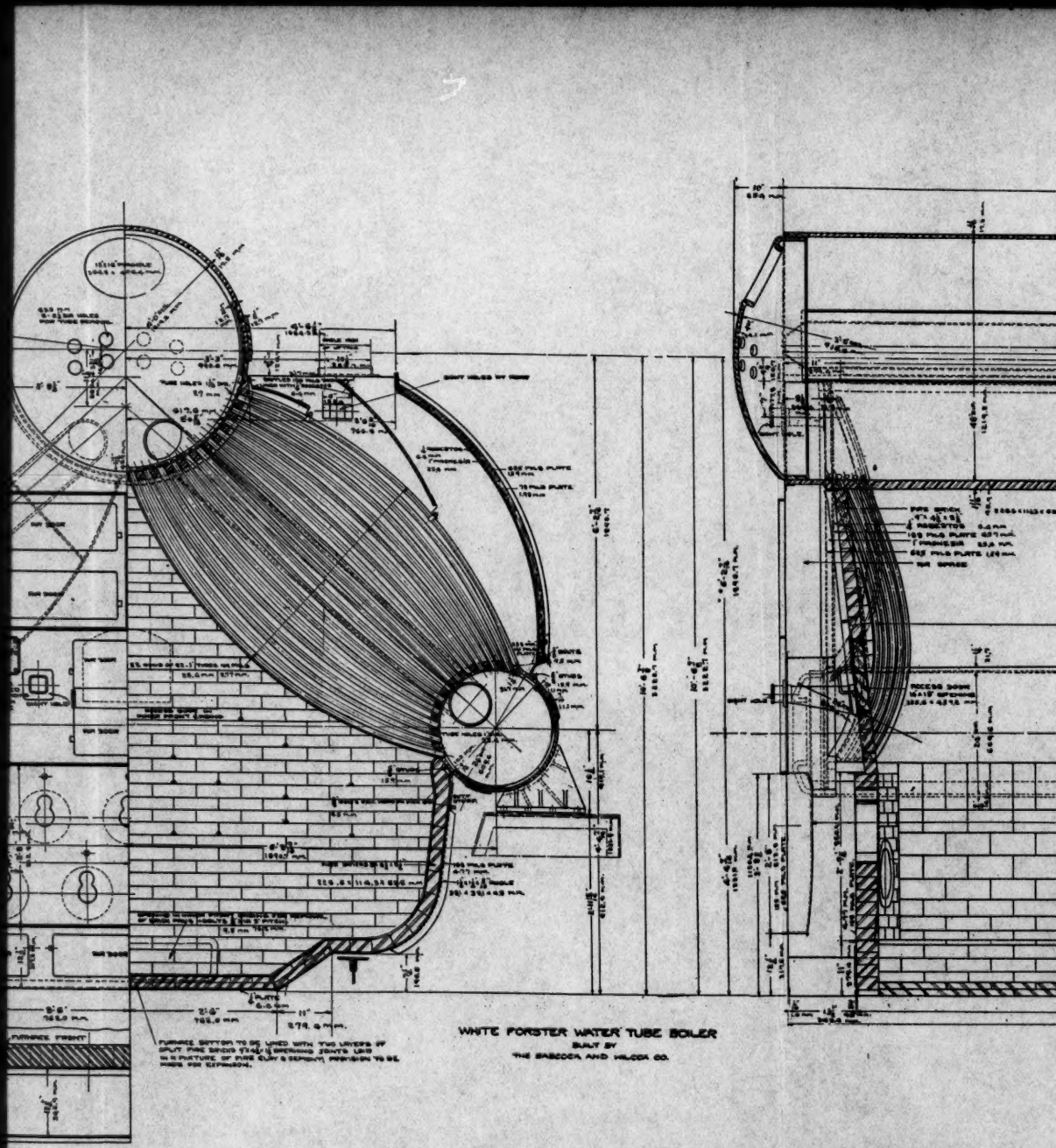
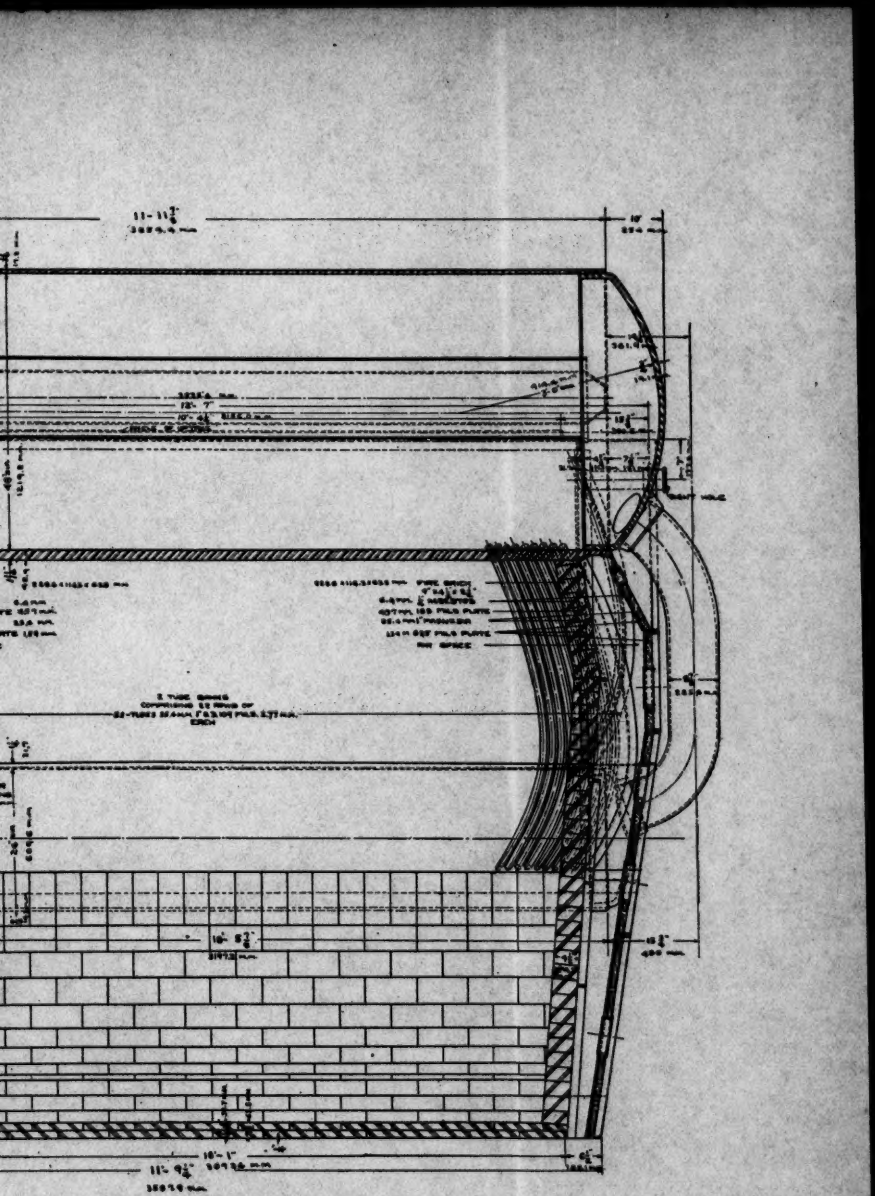

WHITE FORSTER WATER TUBE BOILER
BUILT BY
THE BABCOCK AND WILCOX CO.

FIG. 5.



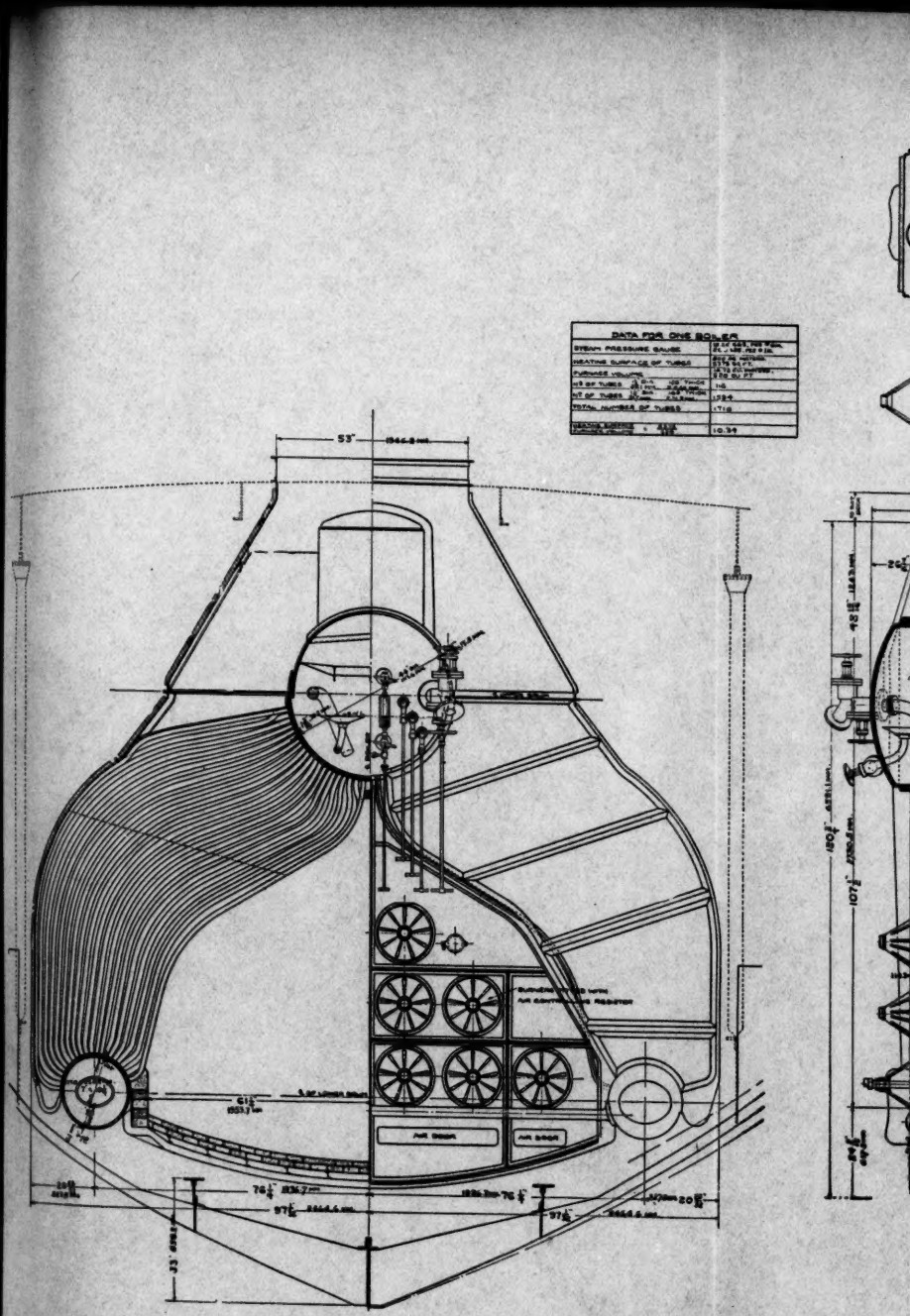
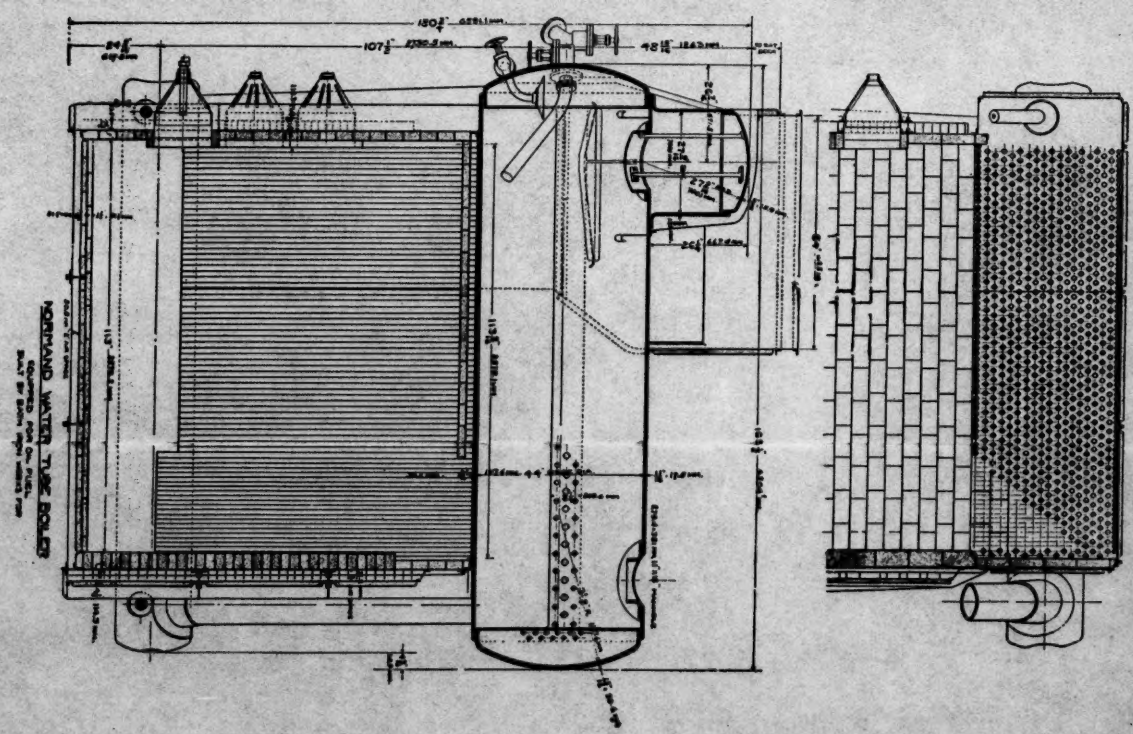

DATA FOR ONE BOILER
STEAM PRESSURE GAUGE
HEATING SURFACE OF TUBES
FURNACE VOLUME
NO OF TUBES
NO OF TUBES
TOTAL NUMBER OF TUBES
53"
AIR DOOR
AIR DOOR

FIG. 6.


NORMAND WATER TUBE BOILER

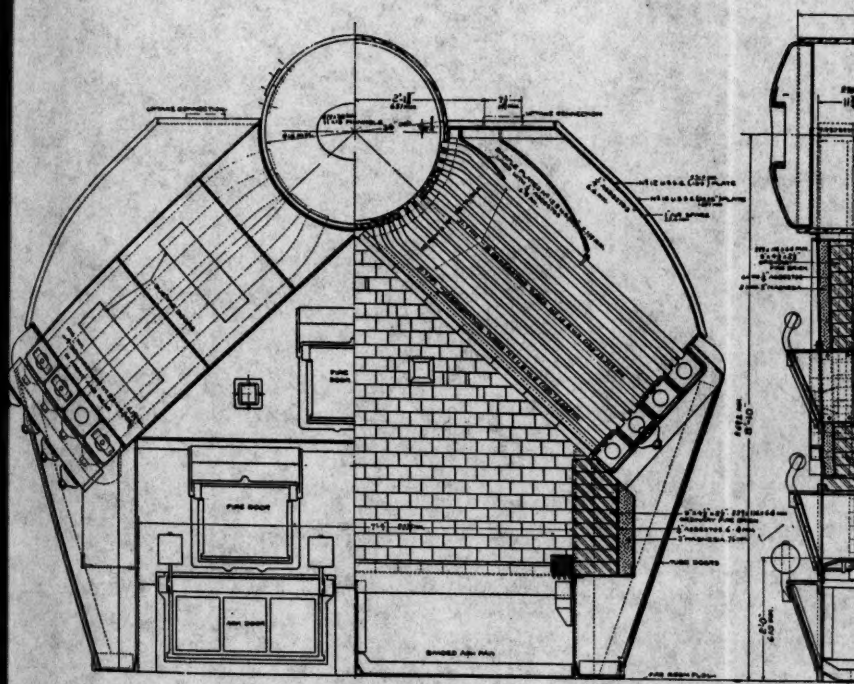
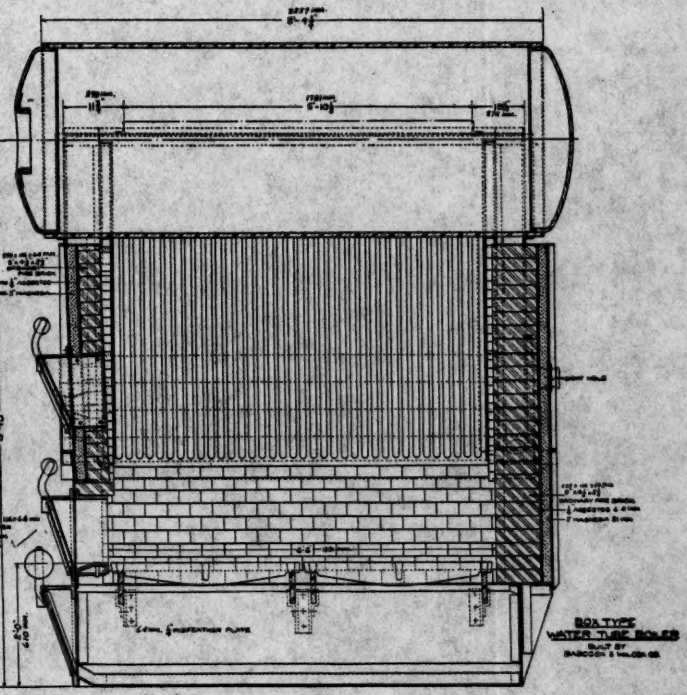
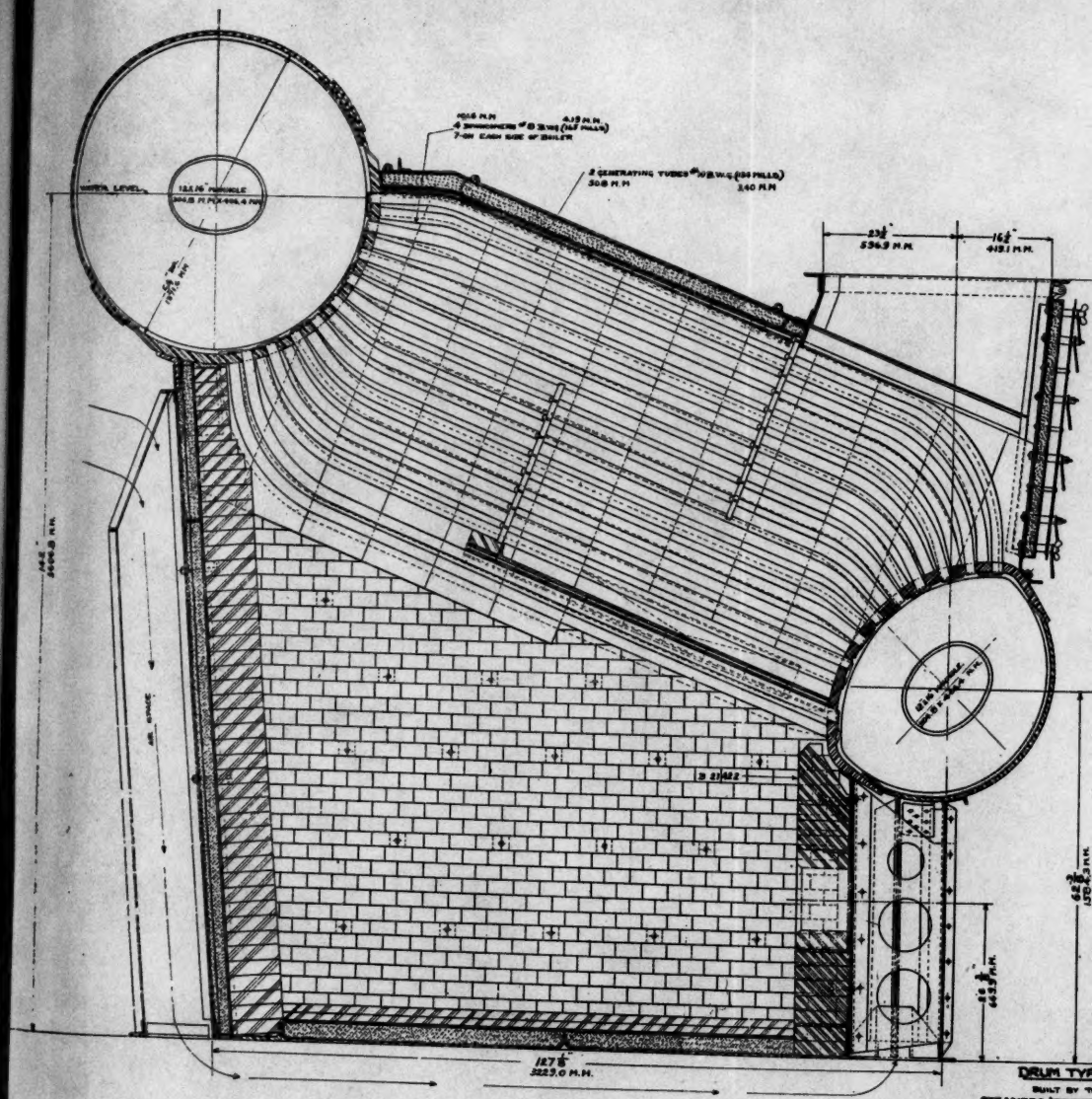


FIG. 7.


BOX TYPE
WATER TUBE BOILER



DRUM TY

BUILT BY

STEAMERS "GREAT

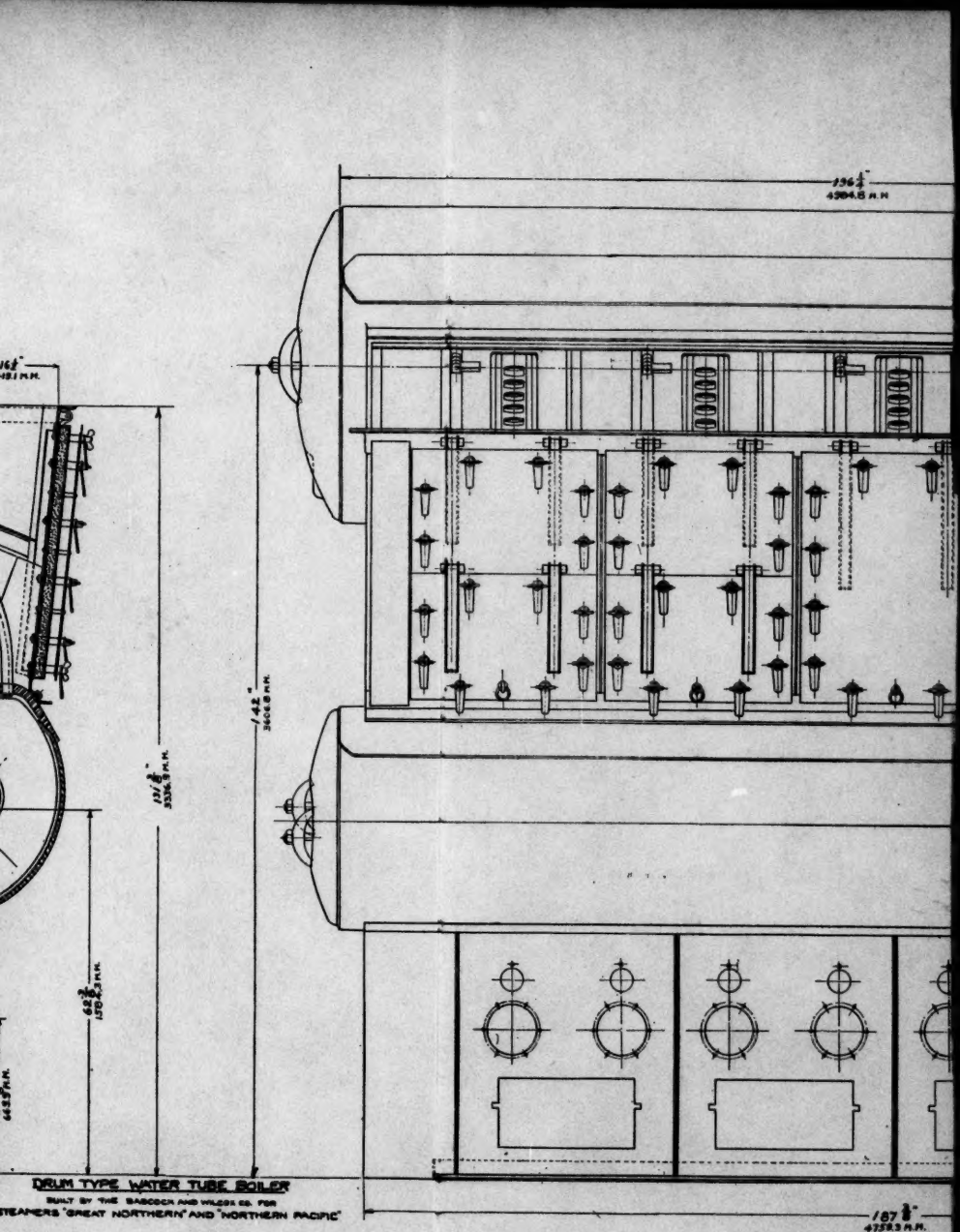

DRUM TYPE WATER TUBE BOILER
BUILT BY THE BABCOCK AND WILCOX CO. FOR
STEAMERS "GREAT NORTHERN" AND "NORTHERN PACIFIC"

FIG. 8.

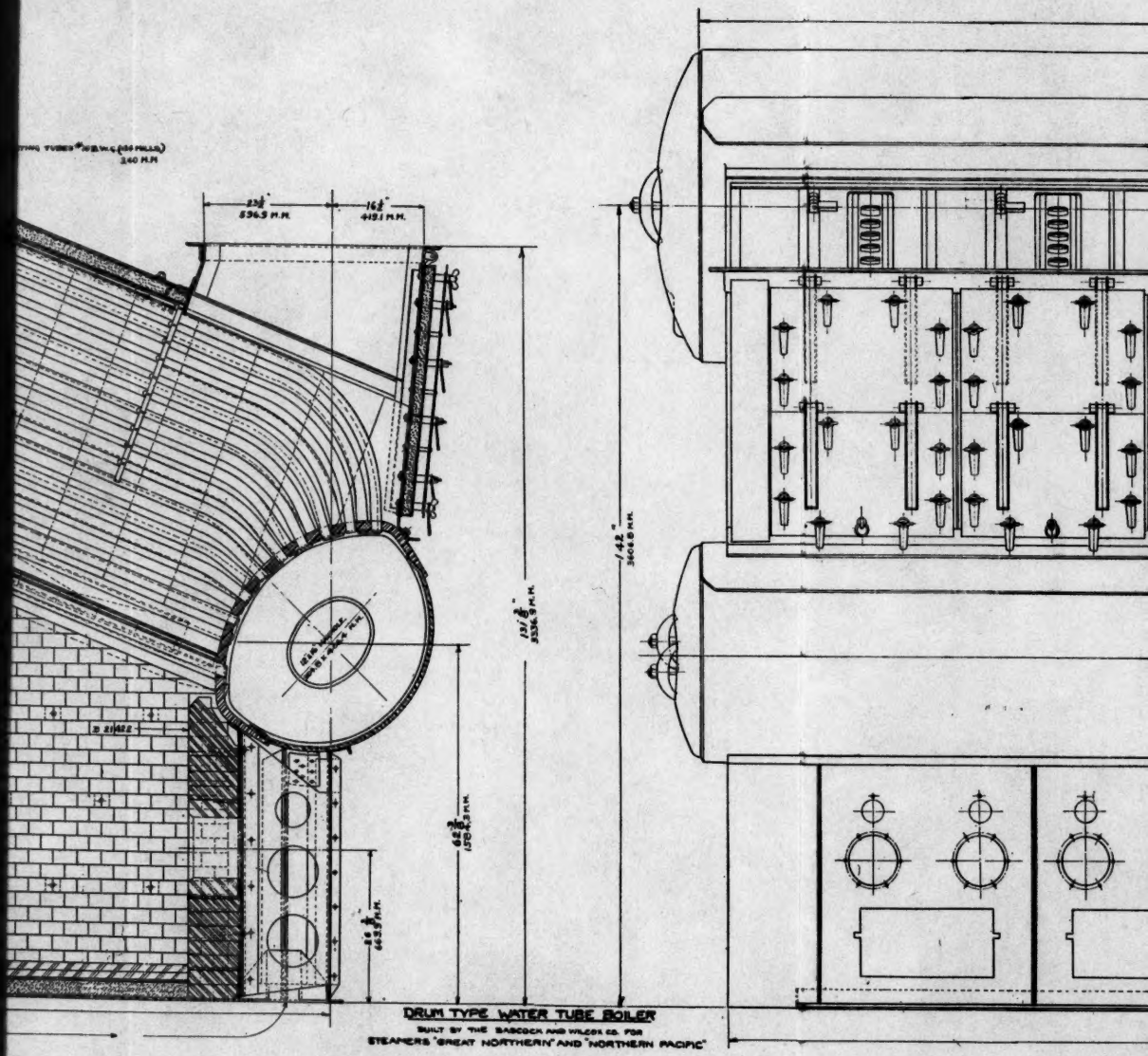
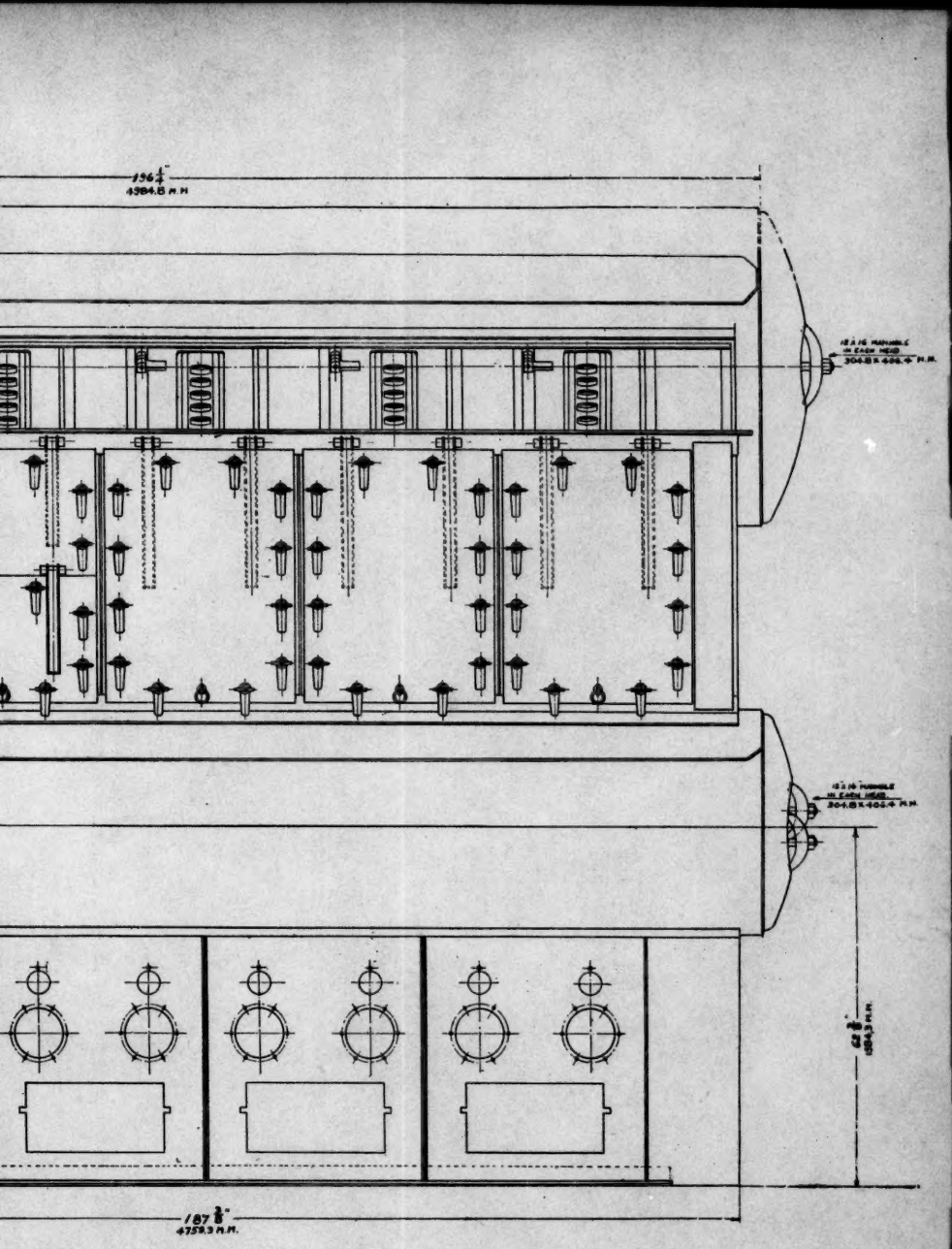

DRUM TYPE WATER TUBE BOILER
BUILT BY THE BABCOCK AND WILCOX CO. FOR
STEAMERS "GREAT NORTHERN" AND "NORTHERN PACIFIC"

FIG. 8.


196 1/4"
4984.8 M.M.
187 3/8"
4759.3 M.M.

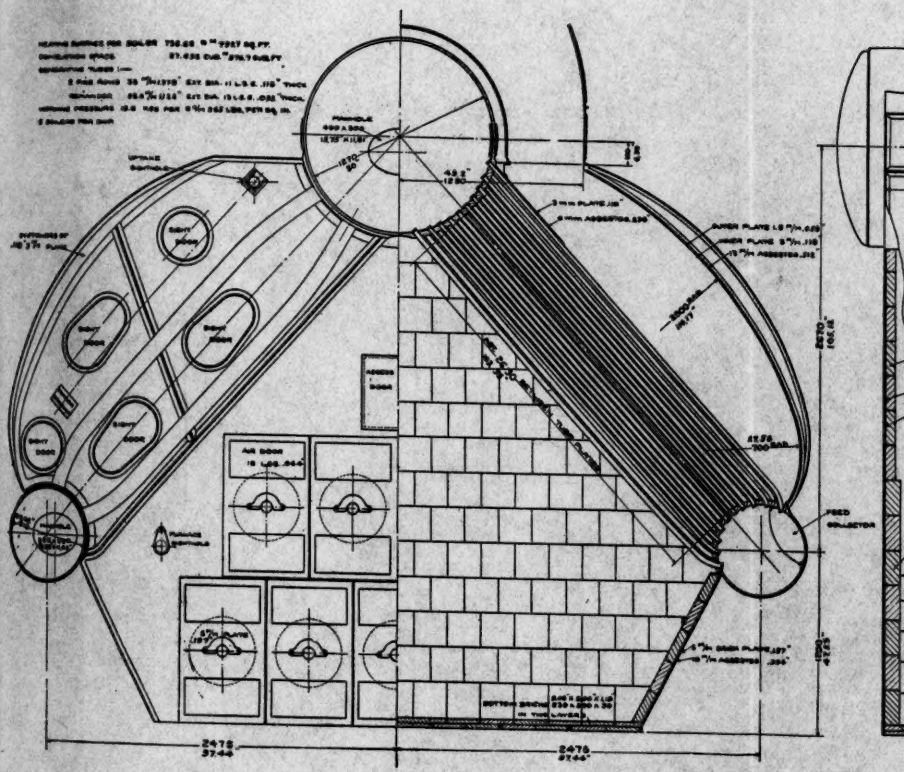
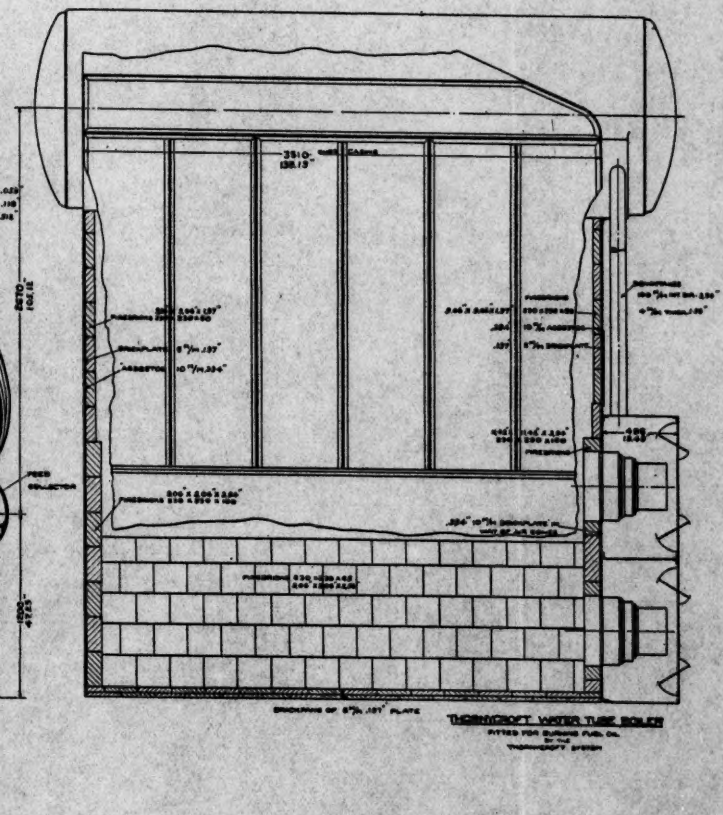


FIG. 9.


THORNYCROFT WATER TUBE BOILER

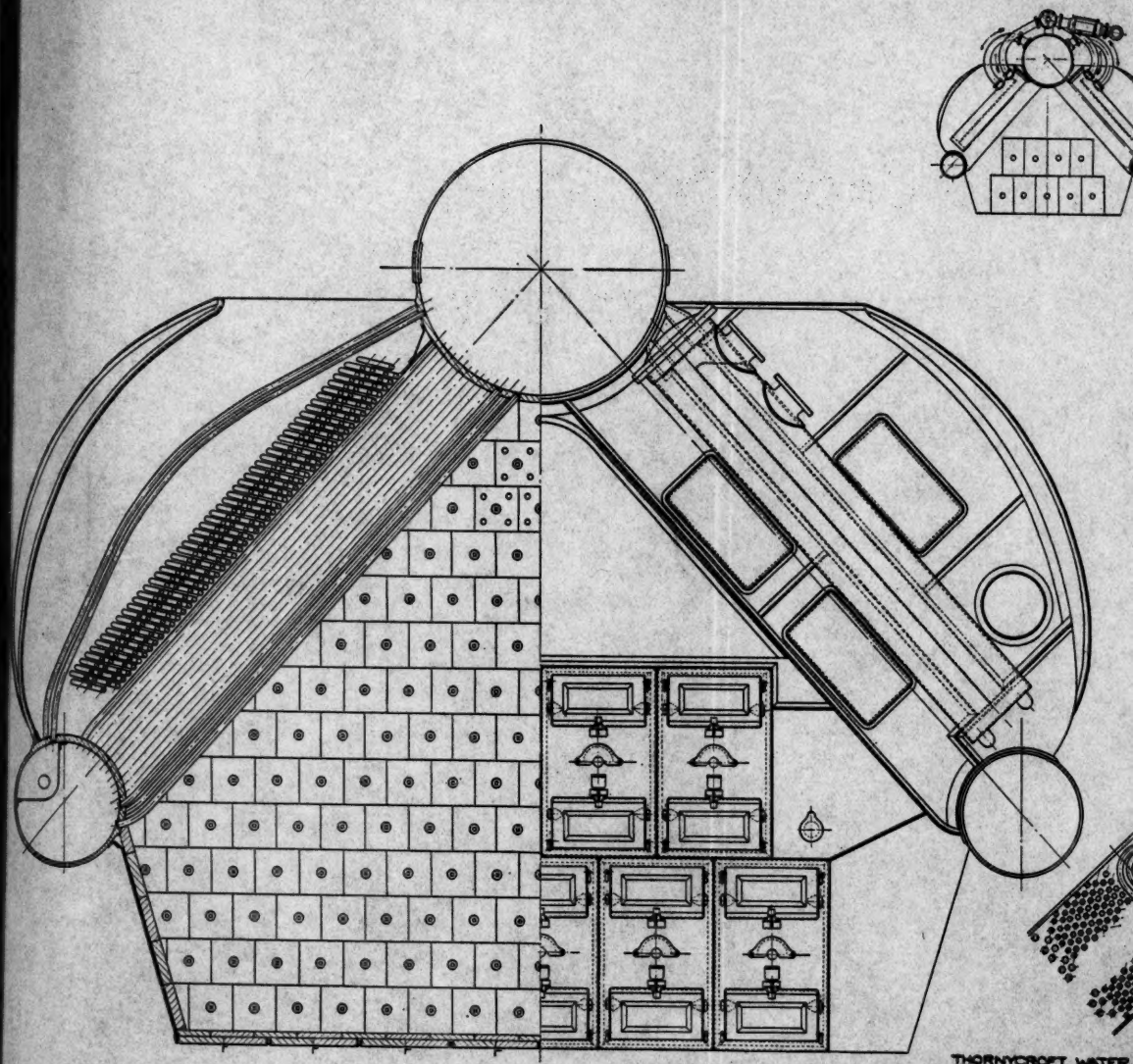


FIG. 10.

OFT WATER TUBE BOILER
FITTED WITH
HEATER AND SUPERHEATER

IG. 10.

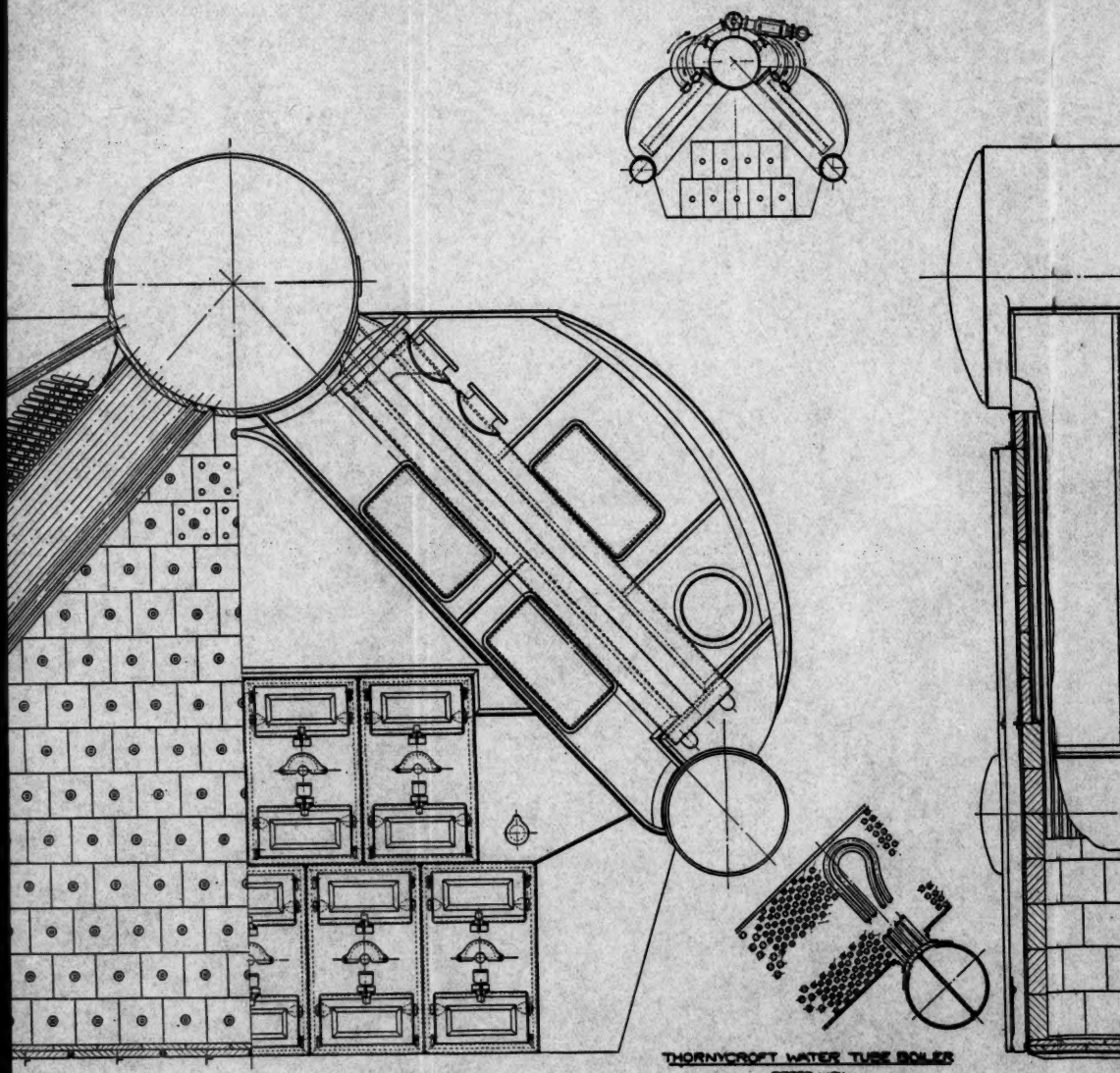
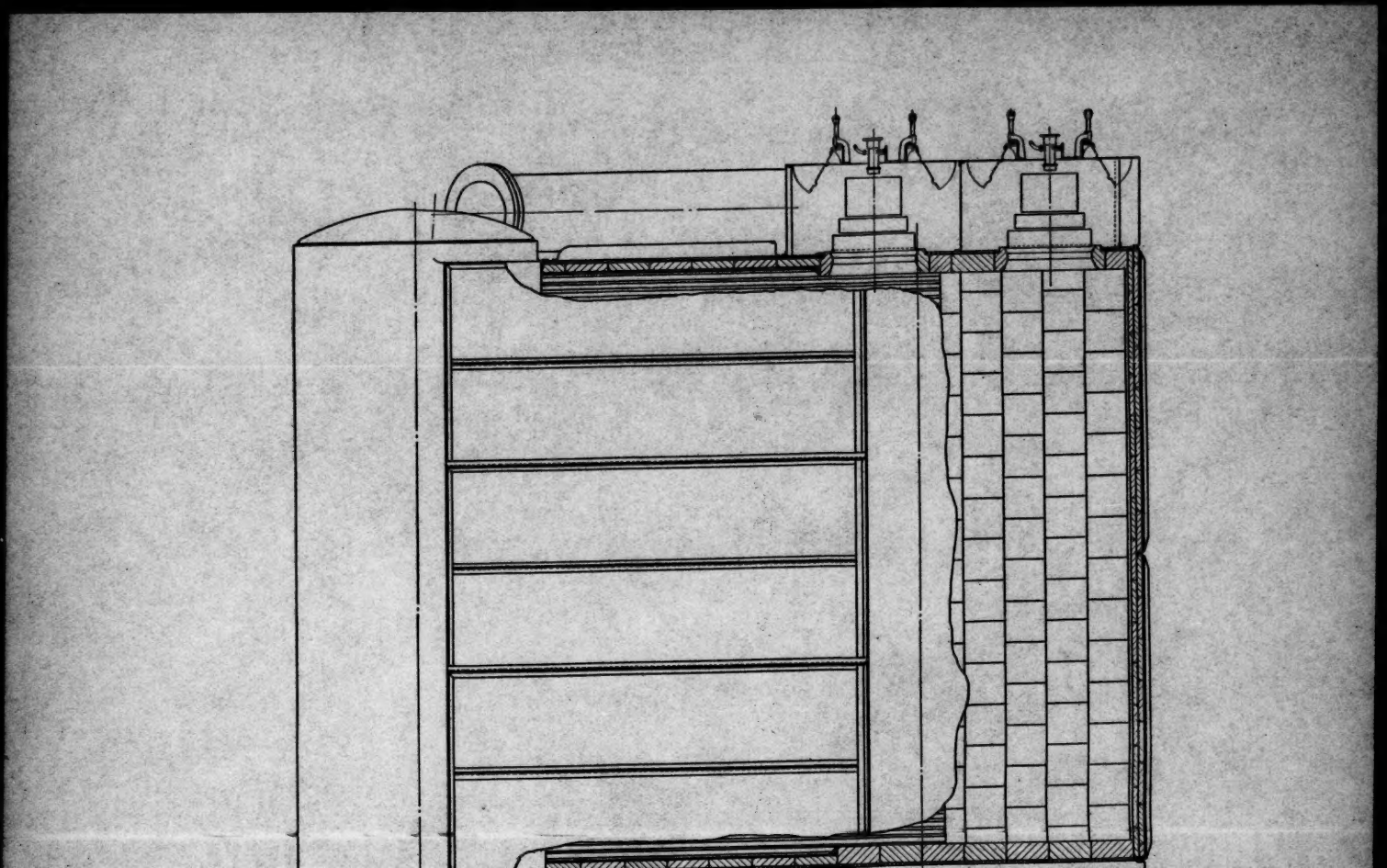
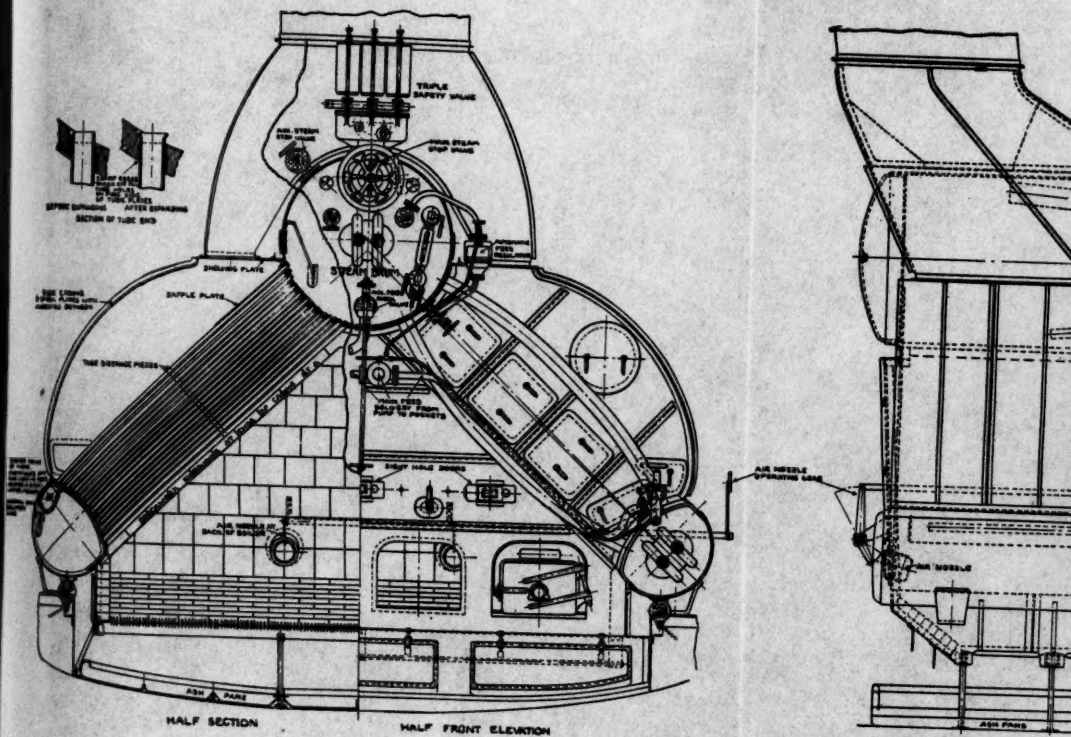

THORNYCROFT WATER TUBE BOILER
FITTED WITH
FEED HEATER AND SUPERHEATER

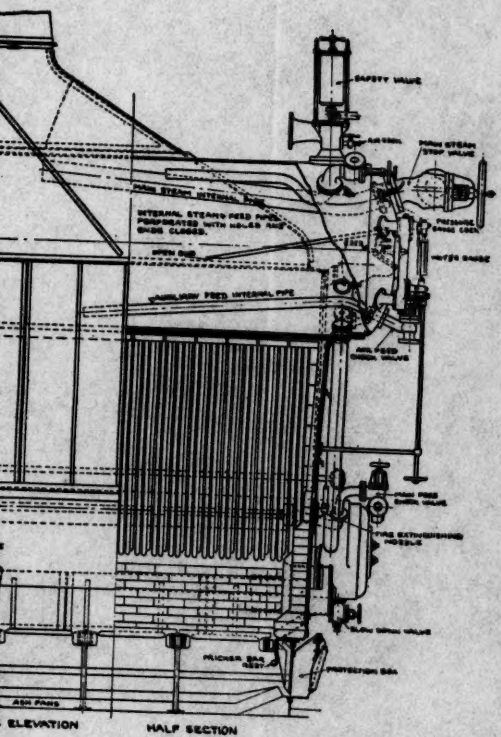
FIG. 10.

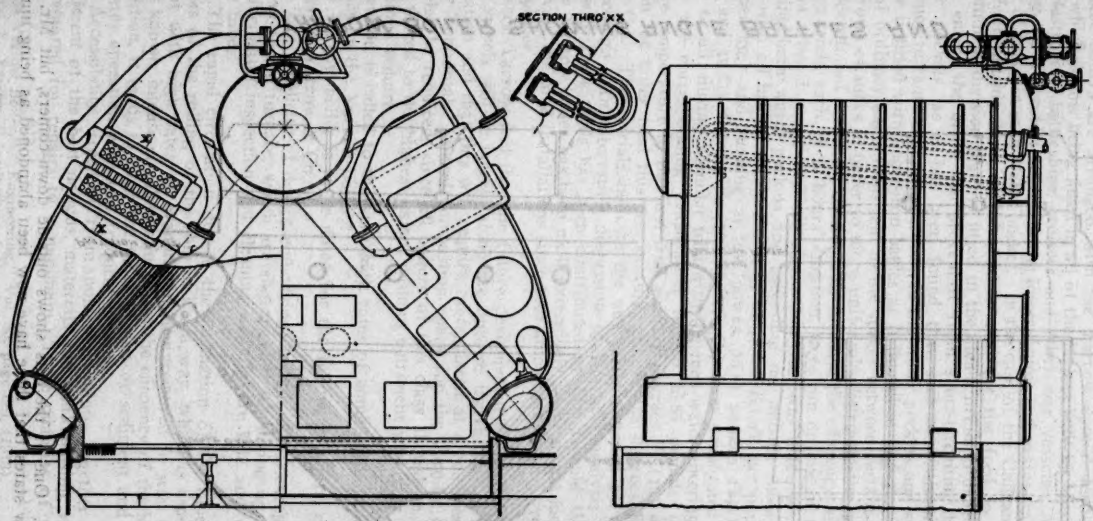




YARROW PATENT WATER TUBE BOILER
FITTED FOR BURNING COAL

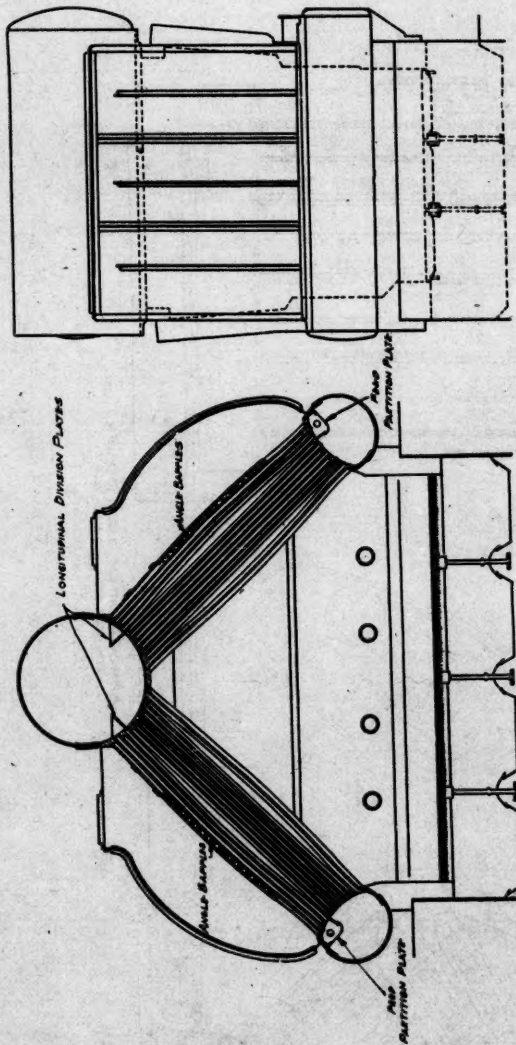
FIG. II.


SAFETY VALVE
INTERNAL STEAM FEED PIPE
PERFORATED WITH HOLES
ENDS CLOSED.
PRESSURE GAUGE
WATER GAUGE
AUXILIARY FEED INTERNAL PIPE
ASH FEED DRAIN VALVE
MAIN FEED CHECK VALVE
FIRE EXTINGUISHER NOZZLES
BLOW DOWN VALVE
PRICHER BAR
PROTECTION BAR
ASH PANS
ELEVATION
HALF SECTION



YARROW WATER TUBE BOILER
WITH STEAM SUPERHEATER
FOR
H.M.S. "YARMOUTH"

FIG. 12.



YARROW BOILER SHOWING ANGLE BAFFLES AND LONGITUDINAL DIVISION PLATES.

FIG. 13.

particulars. One of the cuts shows outside down-comers, but Mr. Harold Yarrow states that these have now been abandoned as being unnecessary.

Messrs. Yarrow state that the forty-two boilers built for the Chilean Battleships *Almirante Latorre* and *Almirante Cochrane* are the last battleship boilers which they have constructed without superheaters.

These boilers are arranged with the Yarrow special feed-heating device which is now being fitted in all of the boilers built by Messrs. Yarrow for destroyers, battleships, cruisers and other war vessels. As indicated in Fig. 13,* the cold feed enters the lower drums and passes up the outside rows of tubes in each water pocket. In the case of the Chilean battleships the four outside rows of tubes were used for the feed to ascend. A longitudinal division plate is fitted in the steam drum to deflect this feed as it enters the drum and prevent it from short-circuiting back to the water pockets, as it has been found that such short-circuiting has a tendency to cause excessive stresses in the water pockets and seams. The boiler is fitted with angle-iron baffles as indicated in Fig. 13. These consist of ordinary angle irons laid in the spaces between certain of the outside rows of tubes in order to more uniformly distribute the gases, and it has been found that these considerably increase the efficiency of the boiler.

Two trials, each of 12 hours' duration, were run on one of the Chilean boilers when burning fuel oil only. One trial was made with the feed entering both water pockets and ascending the outside tubes, on which trial, with oil having a calorific value of 19,000 B.t.u's. per pound (10,550 cal. per kg.) and running at low power, 16.81 pounds (kg.) of water from and at 212 degrees F. (100 degrees C.) were evaporated per pound (kg.) of oil; during a similar trial with the feed water entering the steam drum the corresponding evaporation fell to 15.83 pounds (kg.). The uptake temperature in the first trial was 312 degrees F. (155 degrees C.) and in the latter trial this temperature rose to 423 degrees F. (217 degrees C.). These trials indicate the advantage obtained by this system of feeding.† An interesting discussion of the heat analysis in connection with these trials has been contributed by Mr. Donald W. Rennie.‡

The Yarrow boiler has been most extensively adopted in a large number of navies, the total installations since 1905 amounting to well over 4,000,000 horsepower.

Messrs. Yarrow feel that the future development of marine boilers for naval vessels will be largely along the lines of increasing the rapidity of circulation and that this will require tubes placed at a considerable angle to the horizontal. Liberal combustion space for the burning of oil fuel will also be required, since all boilers of this class should be capable of being forced to the highest extent without detriment whenever emergency conditions require.

Attention is called to the very high rates of combustion which may be obtained in the Yarrow boiler.*

The Yarrow boiler is to be noted for the following points, emphasized by Messrs. Yarrow & Co.

(1) The simplicity of design embodying straight tubes, excepting those adjacent to the furnace, which are slightly bent for taking up expansion. This facilitates inspection and cleaning of both fire side and water side of the tubes.

(2) The rapid circulation due to the inclination of the tubes and regardless of the rolling of the ship in a sea way, which is especially important with high rates of evaporation now required in war ships. As is well known, intense circulation increases the efficiency of the boiler, and, in addition, it tends to clear the tubes of dirt or sediment and therefore facilitates operation with less frequent cleaning.

(3) Accessibility, freedom from numerous joints, together with the embodiment of the best designs, materials and workmanship, reduce the time required for repairs to a minimum. These qualities of the boiler

* "The Yarrow Boiler," "Engineering," Vol. 95, page 681 (1913).
† Harold E. Yarrow, "Institution of Naval Architects," March (1912).
‡ "The Heat Analysis of an Oil-fired Water-tube Boiler," Donald W. Rennie, "The Engineer," Vol. XCVII (1914).

also facilitate rapidity of raising steam, which has been especially emphasized during the last few months.

Babcock & Wilcox Marine Boilers.—This boiler has been installed in both small and large powers in a great number and variety of vessels. It is one of the boilers selected by the British Admiralty after the report by its boiler committee in 1902. The company have large shops in the United States and Scotland equipped with the best special machines and tools, and they also have works in France and Germany.

The dry weight of the boiler when built for 250 to 295 pounds working pressure (17 to 20 atmos.) is about 20 pounds per square foot of heating surface (98 kgs. per sq. m.), as compared with 40 to 50 pounds per square foot (195 to 244 kgs. per sq. m.) for Scotch boilers for 180 pounds pressure (12.2 atmos.). The water in such Scotch boilers varies from 17 to 20 pounds per square foot of heating surface (83 to 98 kgs. per sq. m.). In large Babcock & Wilcox boilers arranged for oil firing and fitted with 2-inch (51 mm.) tubes, No. 10 B.W.G., in thickness, the dry weight is below 16 pounds per square foot of heating surface (78 kgs. per sq. m.).

The water in the boiler amounts to 3 to 5 pounds per square foot (15 to 24 kgs. per sq. m.) of heating surface.

The American coal-burning boiler is constructed with the tubes placed at 15 degrees inclination to the horizontal and with forged-steel side water pockets below the tubes, which form excellent furnace sides for coal firing and efficient heating surface. In the English type of this boiler for coal burning the grates of adjacent boilers sometimes extend to a common division wall between the boilers, with cleaning space provided at the side doors above this division wall; this permits a considerable increase in the grate area, which is an advantage for long high-powered runs where cleaning fires is an important condition.

The American type of boiler, built for oil fuel, is also fitted with side brickwork and without side water boxes below the tubes. The lower rows of tubes immediately above the furnace for oil burning are usually inclined 18 degrees from the horizontal while the tubes above these are inclined 15 degrees. This is shown in Fig. 14.

Reference to Figs. 15, 16, 17 and 21 indicates the accessibility for cleaning and repairs on both the water and fire sides of the tubes. The fire side of the boiler can be cleaned efficiently while under steam, as the cleaning doors are arranged on the side. The tubes, being straight and accessible from either end, are easily inspected and renewed. The Southern Pacific Company find that in their Steamer *Creole* they can blow down a boiler, renew a tube and again get up steam in about two hours.

Admiral Melville's requirement that no cast metal subjected to pressure should be employed in a water-tube boiler has been fully met. The tools for working and forming the pressure parts of these boilers are such as to preserve the best qualities of the materials used, so that a high factor of safety is maintained.

Repeated authentic tests show that steam can be raised from water of 100 degrees F. (38 degrees C.) to a pressure of 200 pounds per square inch (13.6 atmos.) within 15 minutes, without injuring the boiler.

The efficiency of the boiler with oil fuel rises as high as 80 per cent. under test conditions; with coal fuel the corresponding efficiency is about 75 per cent. These efficiencies, of course, are with low rates of combustion, but the efficiency is well maintained with high rates of firing, as will be noted by reference to the tests of the *Arkansas* and *Wyoming* boilers both with coal and oil fuel.* The boiler has been tested to extremely high rates of combustion, corresponding to 70 pounds of coal

* "Test of Babcock & Wilcox Boilers for U. S. Battleships *Wyoming* and *Arkansas*," "Journal of A. S. N. E.," Nov., 1910, Vol. XXII; May, 1911, Vol. XXIII.

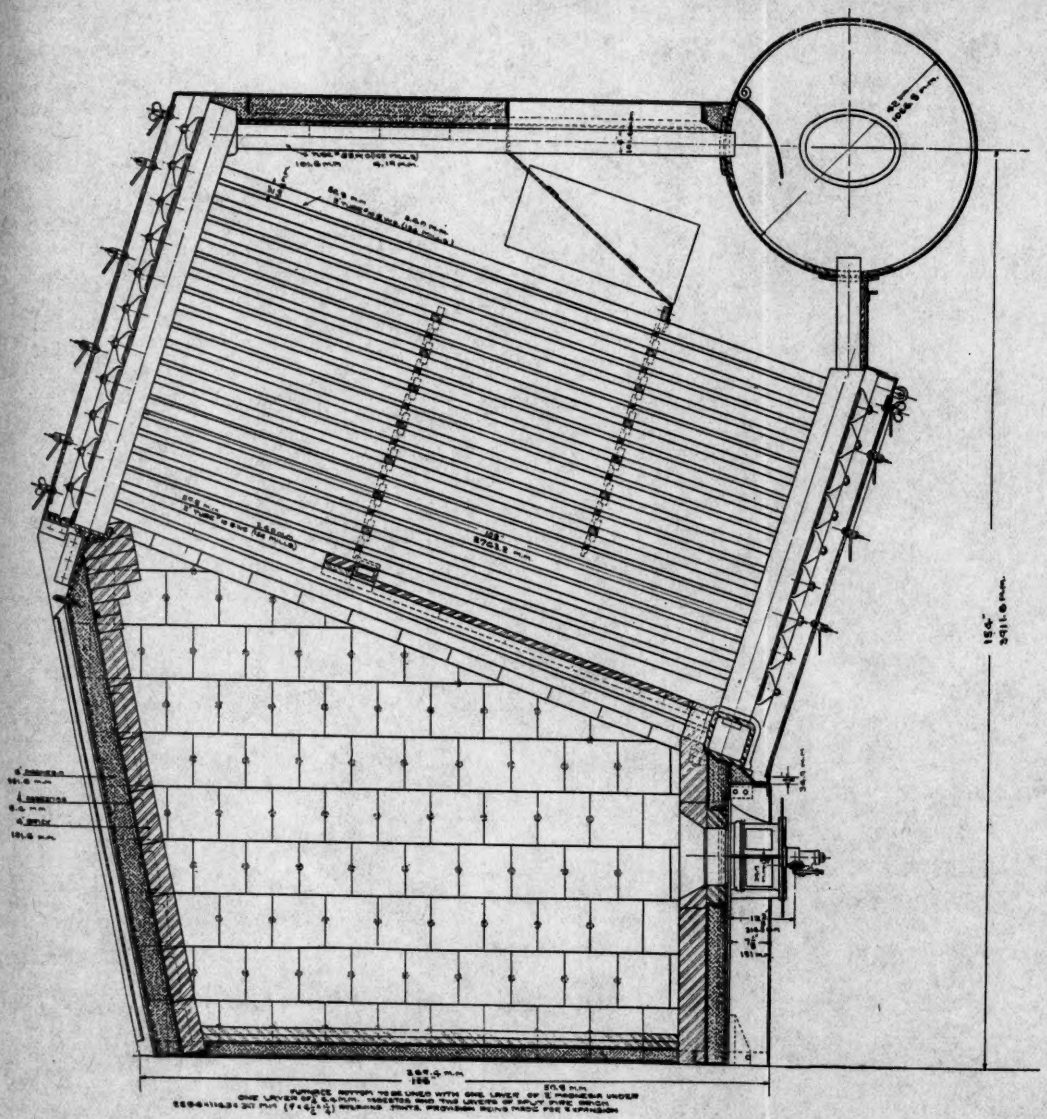


FIG.

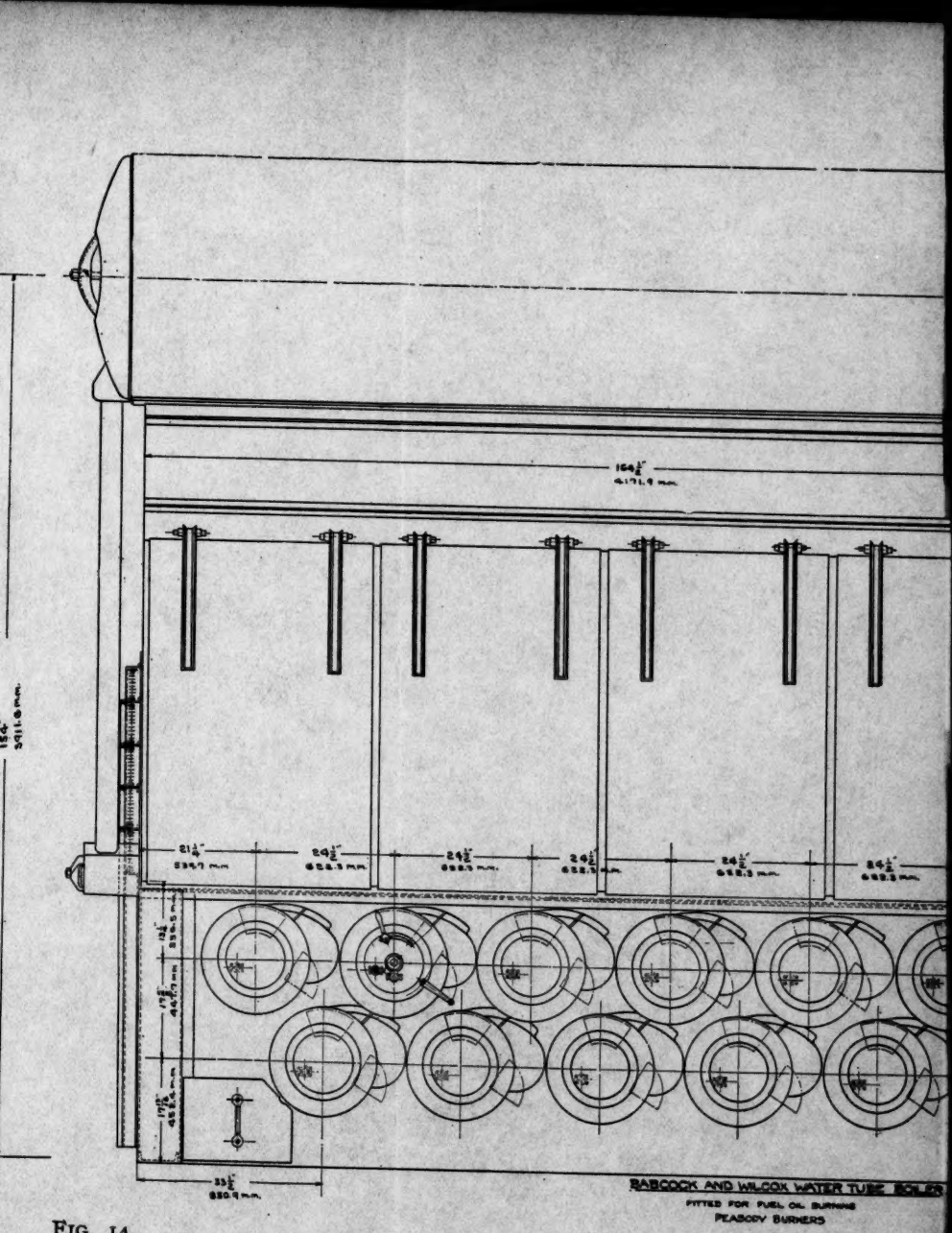

BABCOCK AND WILCOX WATER TUBE BOILER
FITTED FOR FUEL OIL BURNING
PEABODY BURNERS

FIG. 14.

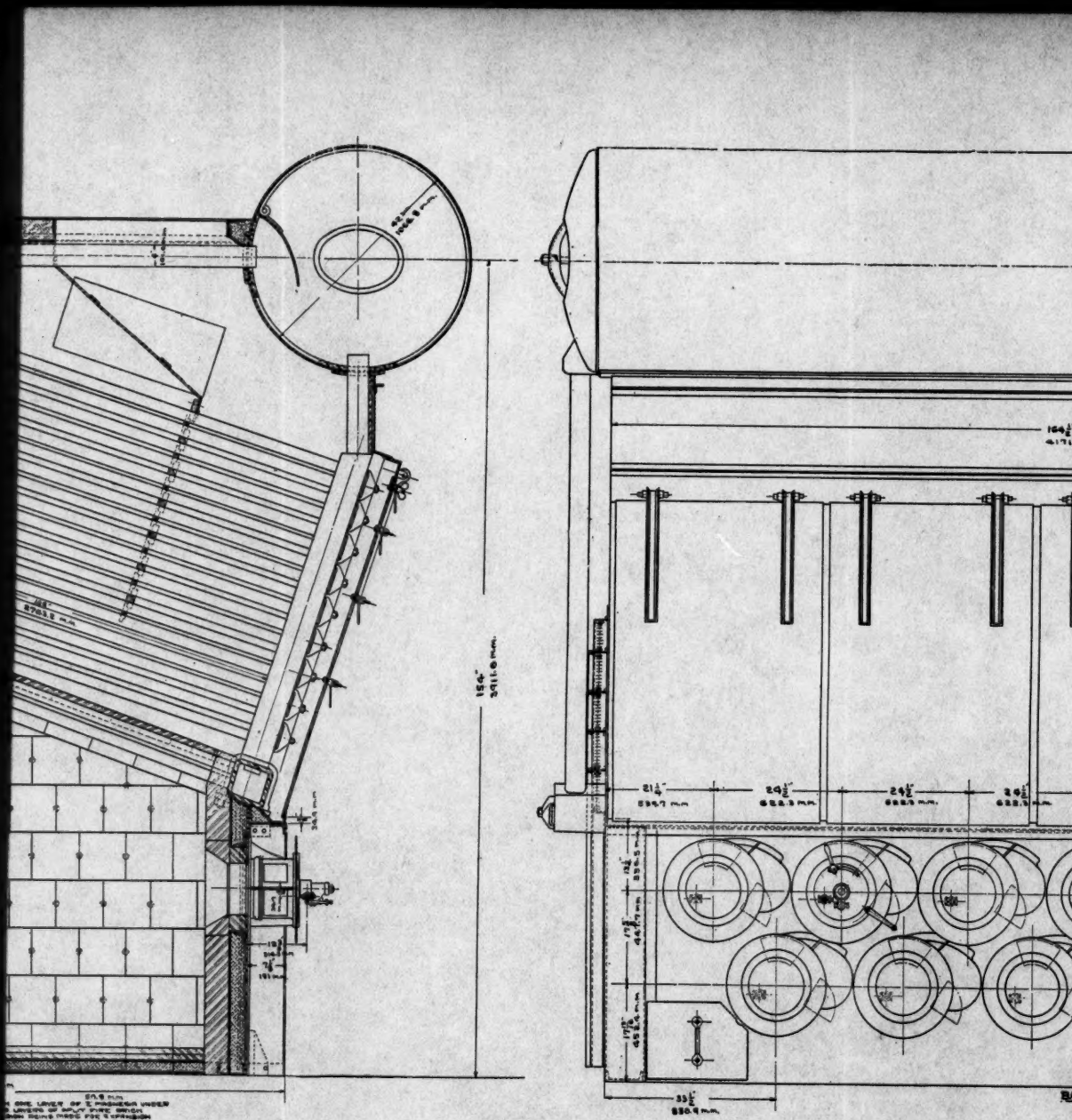
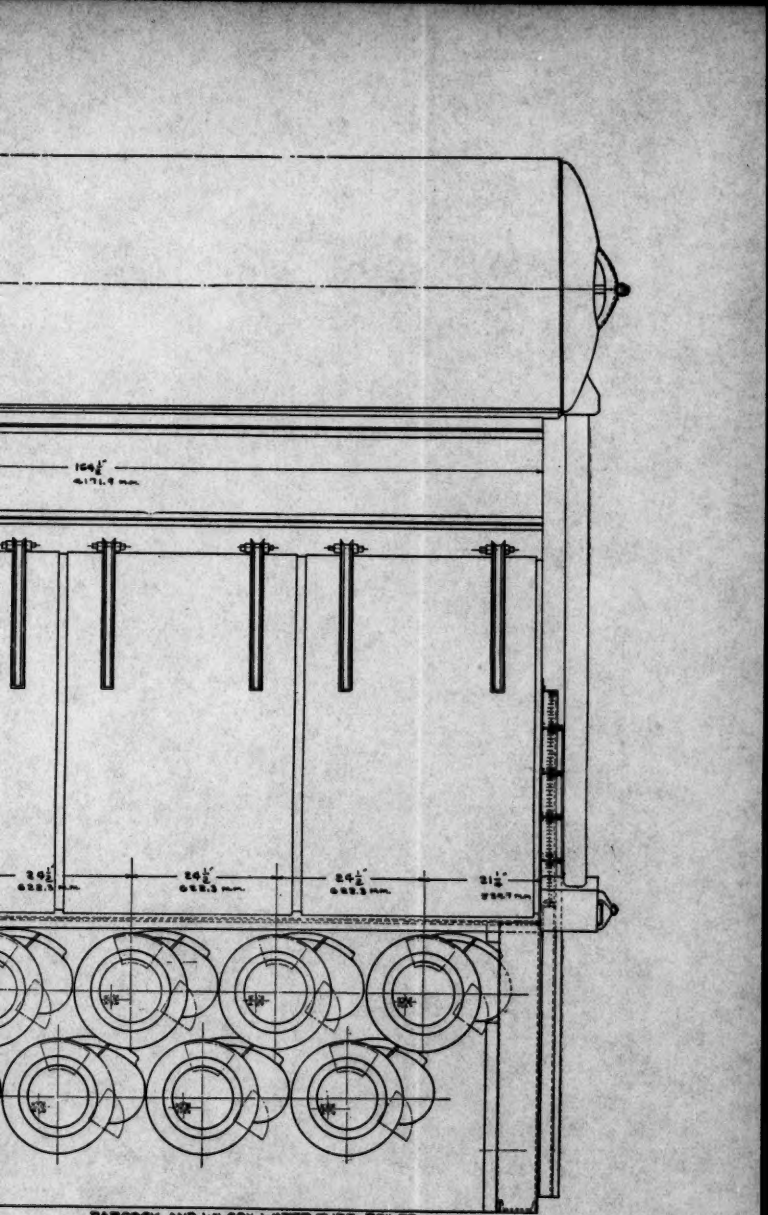


FIG. 14.



BABCOCK AND WILCOX WATER TUBE BOILER

FITTED FOR FUEL OIL BURNING

PEABODY BURNERS

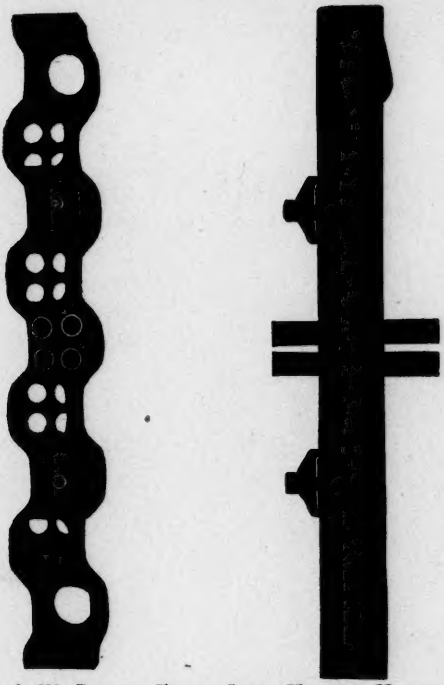


FIG. 15.—B. & W. BOILER. FORGED-STEEL HEADER. HANDHOLE COVERING GROUP OF FOUR 2-INCH TUBES.

FIG. 16.—B. & W. BOILER. DUSTING PANEL—PATENTED.

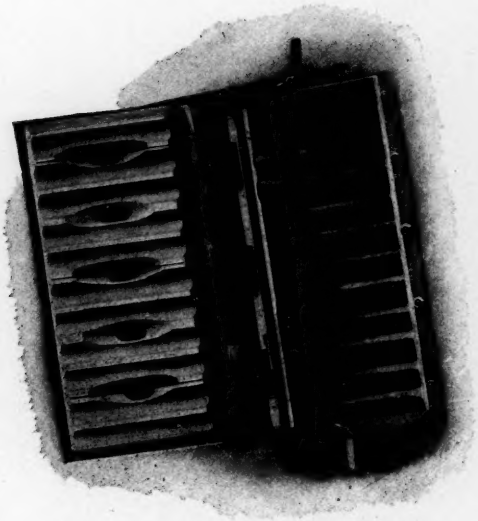


FIG. 17.—B. & W. BOILER. CLEANING PANEL—PATENTED.

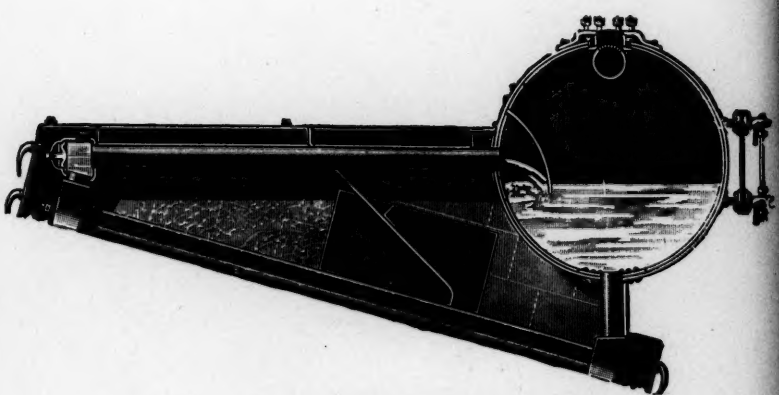


FIG. 18.—B. & W. BOILER. SHOWING DRUM ARRANGEMENT AND CIRCULATION.

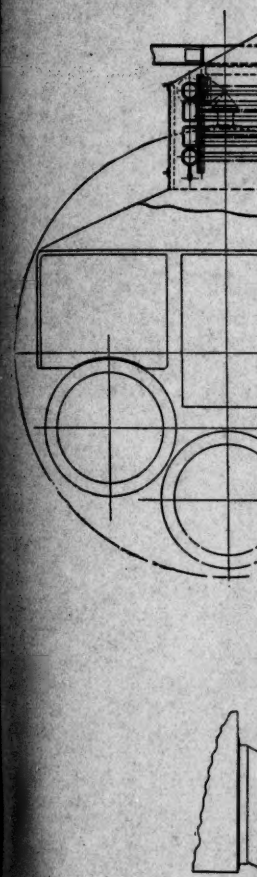
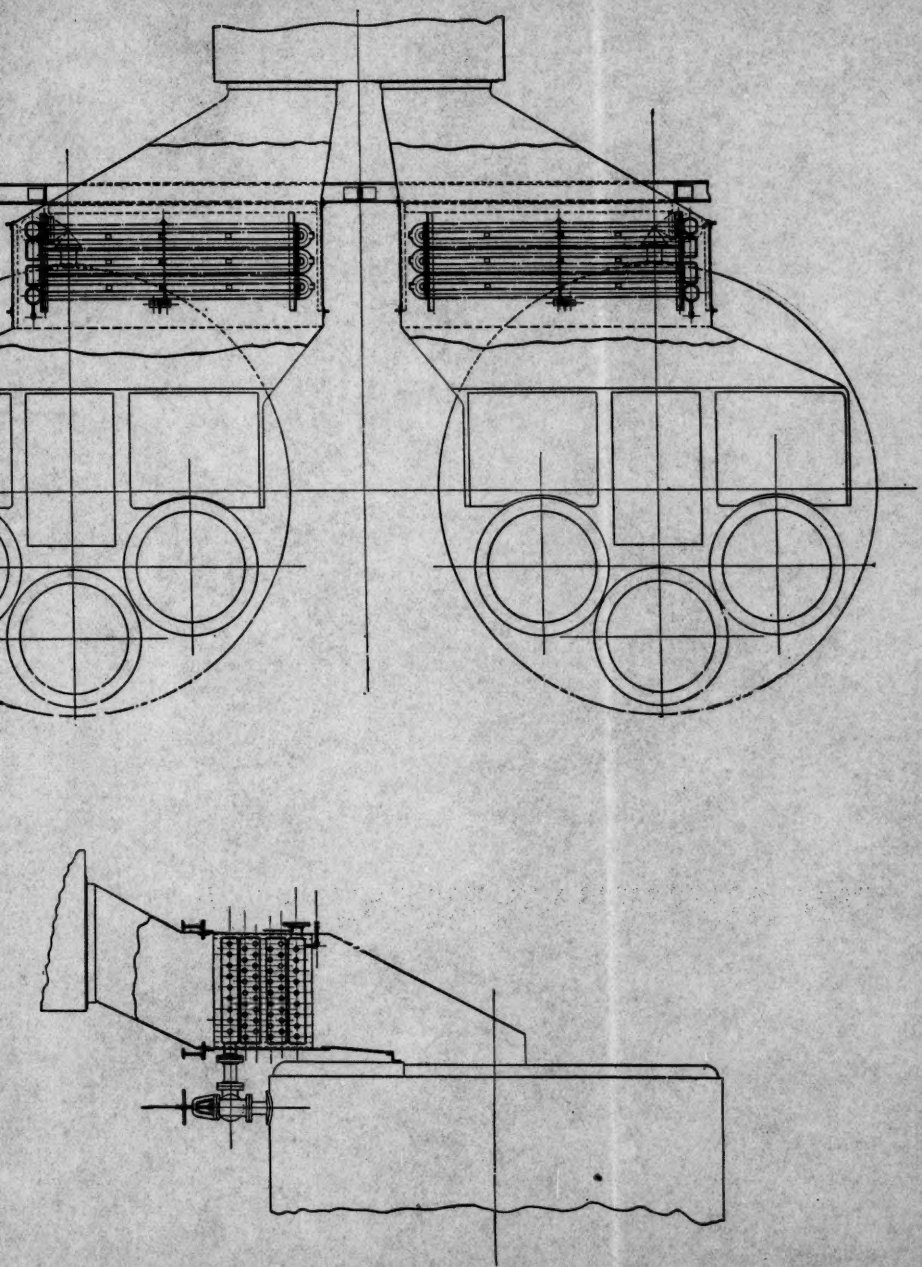


FIG. 19.—ARRANGE



—Arrangement of Foster Superheaters in Uptakes of Steamer "Lyman Stewart."

per square foot of grate surface per hour (342 kgs. per sq. m.), and such tests have been run without injury.

The space occupied by the Babcock & Wilcox boiler is small as compared with Scotch boilers and satisfactory as compared with other water-tube boilers.

The parts of the boiler are made interchangeable and although many are special. still the items subject to the greatest deterioration are of commercial sizes which can be readily procured.

The form of furnace is well adapted to complete combustion before the gases enter the tubes. the vertical baffles direct the gases three times, at right angles, across the tubes, and since the tubes are staggered, as will be seen by reference to Fig. 15 showing the header, the gases are brought into intimate contact with the heating surface.

The circulation in the boiler is rapid and efficient and at the same time the steam surface is liberal and the quality of the steam is remarkably dry. Fig. 18 shows the drum arrangement and circulation. The feed enters the large drum and descends through the nipples and front headers, from which it passes into the tubes; mixed steam and water fill the back headers and pass through the return tubes entering the steam and water drum back of the baffle plate; the steam passes around the ends of the baffle into the steam space. Tests of the *Wyoming* boiler showed that when evaporating 14¼ pounds of water per square foot of heating surface (72 kgs. per sq. m.) from and at 212 degrees F. (100 degrees C.) the steam was 99.57 per cent. dry.

The boiler is of rugged construction and experience shows that it can be efficiently operated by the regular firemen found in service. The furnaces being all of one height is a great advantage and much favored by the firemen, as it is easier to work the fires in such furnaces than to work the high side fires of four-furnace Scotch boilers.

In regard to durability, it may be said that these boilers have been in operation in several steamships for periods of from eight to fourteen years with only minor expenses for repairs and are still in good condition. To cite an example, the steamship *Creole* had ten Babcock & Wilcox boilers installed in 1907 containing 28,500 square feet of heating surface (2,650 sq. m.), 4,350 square feet (404 sq. m.) of superheating surface and 783 square feet (72.7 sq. m.) of grate surface. The trip is usually made operating only seven of the boilers. The pressure parts of the boilers, with the exception of a few superheater tubes, do not show any appreciable deterioration. This condition is attributed to the fact that the boiler water has been kept fresh.

Experience shows that these boilers can be kept clean as easily as Scotch boilers. Boiler cleaning in the *Creole* is all performed by the crew. The owners have sometimes employed outside labor to clean the internal parts of Scotch boilers in other vessels of their line.

The expense for upkeep of these boilers has been very much less than the corresponding expense of upkeep of Scotch boilers in sister vessels. The deterioration of the boilers in use is less than that of the boilers which are not in use. The casings are still in good condition and less expense has been entailed in maintaining these casings than has been expended in repairs to the ordinary galvanized-iron covering over the lagging used on the Scotch boilers of sister vessels.

Much along the same line might be said for this boiler in naval vessels, but in such work the water-tube boiler is a necessity and the grade of men to operate and care for them is higher than in merchant vessels.

SUPERHEATED STEAM.

The saving due to superheat has long been recognized, but marine engineers have been slow to take advantage of this saving owing to the fear

of trouble in the lubrication of the cylinders, the danger of oil in the boilers, and to the effect of superheated steam on the cylinders, valves, pipes and fittings. Difficulty in connection with pipes, fittings and valves has been mostly overcome, and now the use of superheated steam in marine work is rapidly increasing; the adoption of turbines with reduction devices will minimize the danger in connection with superheat used in large direct-connected turbines.

Many installations have demonstrated a great economy by the use of superheated steam. From 8 per cent. to 15 per cent. saving in fuel consumption frequently appears possible. Several forms of marine superheaters are being installed, principally of the U-tube type. The best known types are the Foster, Babcock & Wilcox, Niclausse, Schmidt, Yarrow and the Thornycroft superheaters. The independently-fired superheater is not much used in marine work. Superheaters formed of straight tubes rigidly held at each end appear to be difficult to keep tight.

The Foster Superheater is arranged in the uptakes, and therefore is not adapted for high degrees of superheat, but with ordinary boiler pressures of about 180 pounds per square inch (12.2 atmos.) it is feasible to obtain from 60 degrees to 65 degrees F. superheat (15 degrees to 18 degrees C.). This gives the advantage of thoroughly dry steam of temperatures which are not too high for use in the main engines and auxiliaries without the use of special lubrication, and also facilitates the maneuvering and starting operations.

This system is made up of a system of coils of U-tubes, and the steam is conducted in parallel, the run being about 50 feet (15.2 m.). The steam is passed back and forth six or more times at a velocity of about 5,000 feet per minute (1,525 m.). The tubes are usually two inches diameter, of cold-drawn steel bent to U-shape and connected through forged-steel headers. The exterior of the tubes is covered with cast-iron casings accurately bored and tightly shrunk on.

Fig. 19 shows the arrangement of the Foster superheater in the uptakes of the steamer *Lyman Stewart,* which is plying on the Pacific Coast.

Fig. 20 shows the method of connecting the U-tubes to the headers and

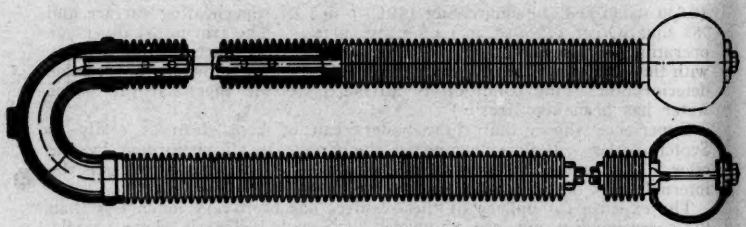


FIG. 20.—CROSS-SECTIONAL VIEW OF RETURN BEND ELEMENT AND CONNECTING HEADERS USED IN THE CONSTRUCTION OF FOSTER SUPERHEATERS.

the cast-iron casing covering the tubes. This cut also indicates the interior tubes which are placed within the superheating tubes, thus providing a narrow annular space for the passage of the steam, which increases the heating efficiency.

The Thornycroft superheater is shown in Fig. 10 and consists of the U-tubes placed between the main body of generating tubes and the casing.

The Babcock & Wilcox Superheater is arranged in the boiler as indicated in Fig. 21. Due to the location of the superheater at the end of the first pass of the gases through the boiler tubes, arrangement of baffles,

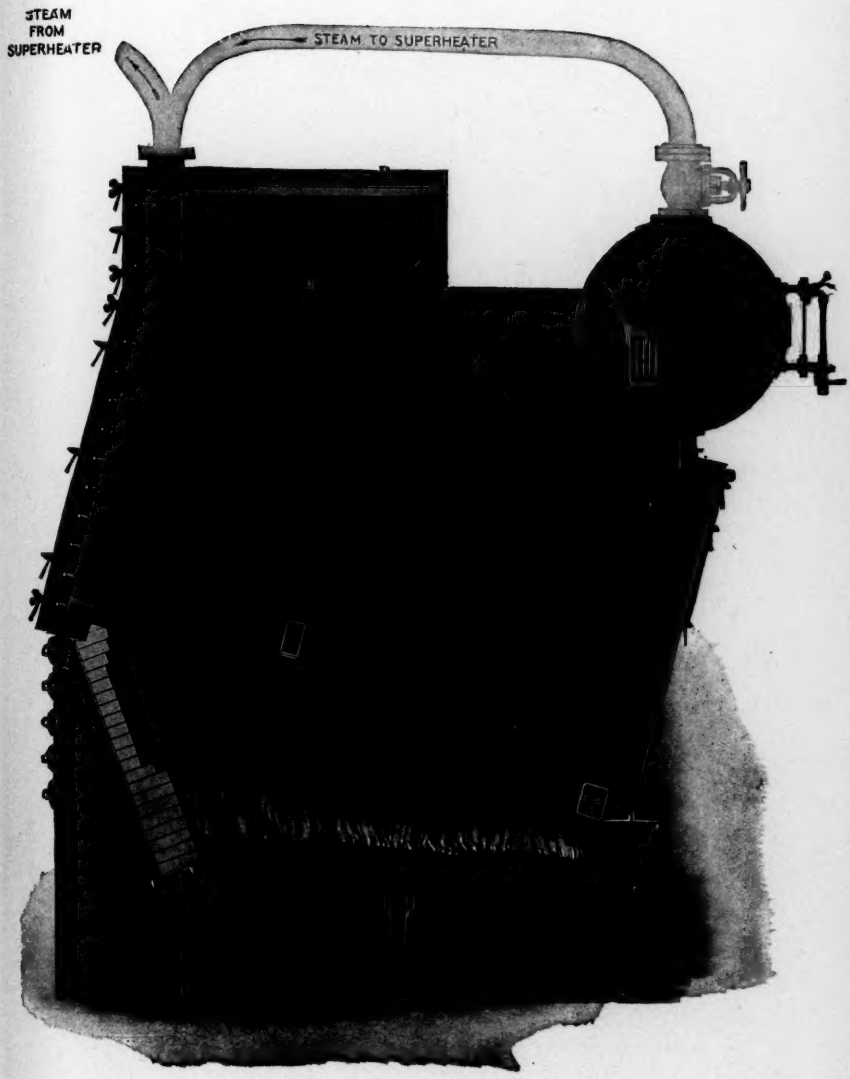


FIG. 21.—BABCOCK & WILCOX SUPERHEATER.

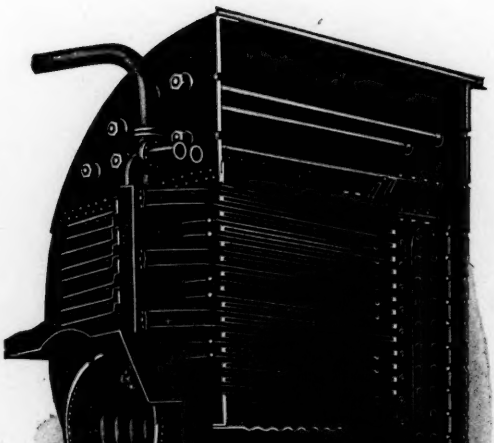


FIG. 22.—SECTIONAL VIEW OF INTERNALLY-FIRED MARINE BOILER, SHOWING LOCATION OF SUPERHEATER ELEMENTS IN FIRE TUBES.

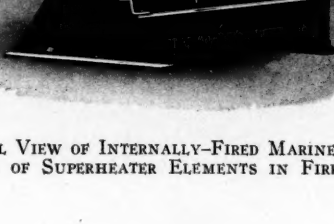


FIG. 23.—SECTIONAL VIEW OF CONTINUOUS RETURN BEND USED IN SCHMIDT FIRE-TUBE SUPERHEATER ELEMENTS.

and the practicability of providing more or less heating surface, a considerable range of superheat may be provided.

These superheaters in service have proven durable and satisfactory, both in merchant work and in naval vessels, with pressures as high as 300 pounds (20.4 atmos.).

The superheaters in connection with the Babcock & Wilcox boilers in the steamer *Creole* which have been in service since 1907 are in good condition, with the exception that some of the superheater tubes show minor corrosion at the ends extending into the headers. In such a case the tube is easily cut off just back of the header and reinserted and expanded with very slight loss of superheating surface.

Yarrow Superheater.—One form of Yarrow superheater is indicated in Fig. 12. Other proposed forms have been illustrated.*

Mr. Yarrow informs the author that all of the war ships they are now building are being fitted with superheaters and there is no doubt that there is advantage in both high and low speeds by their adoption.

The Robinson Type of Superheater was illustrated in April, 1915, in the "Shipbuilding and Shipping Record."

The Robinson superheater is particularly applicable to Scotch boilers. It is of the principle of the small U-tube inserted in the regular heating tubes of the boiler. A unique arrangement of inserting the tubes in the headers is employed.

The Schmidt Superheater has been extensively used in several countries for locomotive service and is also probably the most widely adopted superheater for merchant-marine work. The principle of this superheater is shown in Figs. 22 and 23. It is largely used in connection with Scotch boilers, and, as is well known, consists of the small U-tubes inserted in the regular tubes of the boiler.

Over 1,200 steamers, which total over 1,500,000 horsepower, are now fitted with these superheaters, either as original installations, or as changes from saturated steam jobs. One advantage of this type is that it can be readily installed in saturated-steam Scotch boilers with few changes in piping, it being necessary, of course, to modify the engine valves to adapt them for high temperatures. The temperatures used in marine boilers with this superheater are from 575 degrees F. to 625 degrees F. (302 degrees to 329 degrees C.).

Sir Charles A. Parsons† states in connection with superheaters of the Schmidt type that he considers: "First and most important the placing of the superheater tubes in close proximity to a boiler surface, as when a single convolution of a superheater tube lies within the fire tube of a Scotch boiler, the great advantage being that the superheater tube is saved from burning by the rapidity of its radiation to the contiguous surface; the second important factor is that of high steam velocity so as to scour out and keep the tubes clean."

Calibrated pyrometers should be installed in the engine room, and saturated steam connections should be fitted to the superheated-steam pipes for regulating the amount of superheat. Such an arrangement is shown in Fig. 24.

The U-tubes are connected to the headers of the Schmidt superheater by an ingenious clamp which enables any tube to be quickly removed. Very little difficulty has been found with these connections.

The cleaning of the boiler tubes when fitted with this type of superheater is a point which some engineers may consider serious, but as this superheater is usually installed in boilers fitted with forced draft, and as soot blowers of the Diamond type which may be operated from the backs

* "Results of Experiments with a Water-Tube Boiler with Special Reference to Superheating," Harold E. Yarrow, *Institution of Naval Architects*, 1912.

†North-East Coast Institution of Engineers and Shipbuilders, Volume XXX, "Transaction," 1913-1914.

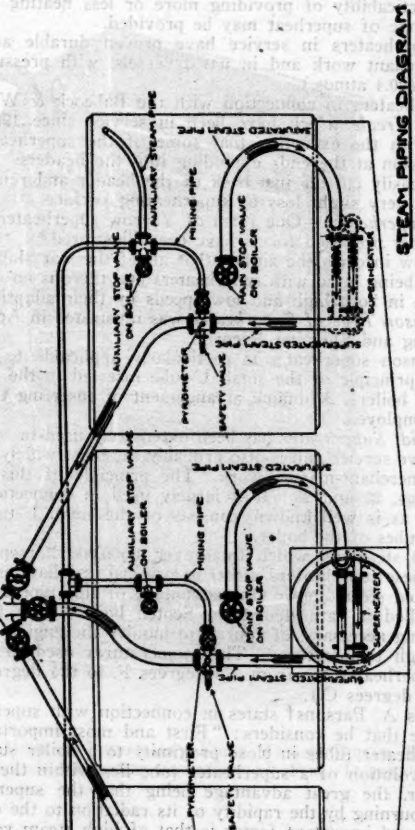

STEAM PIPING DIAGRAM
FOR SCOTCH BOILERS
FITTED WITH
SCHMIDT SUPERHEATERS
ALL PIPES COMING IN CONTACT WITH SUPERHEATED STEAM TO BE
MADE OF SEAMLESS DRAWN STEEL. ALL TEE PIECES AND
STOP VALVES OF CAST STEEL.
SATURATED STEAM PIPE
AUXILIARY STEAM PIPE
MAIN PIPE
AUXILIARY STOP VALVE ON BOILER
MAIN STOP VALVE ON BOILER
SUPERHEATER
SUPERHEATED STEAM PIPE
PYROMETER
SAFETY VALVE
TO THE ENGINE

FIG. 24.

of the boilers are generally fitted in the combustion chambers, the tubes are swept each watch with very little work and kept in efficient condition.

The Diamond blower as fitted in connection with this superheater is illustrated in Figs. 25 and 26. The nozzle of the blower, as it is operated, travels out from its casing which protects it from the heat. As it travels out and in the angle of the nozzle changes so as to direct the steam or air, which is used for blowing the soot, in an increasing and decreasing spiral which reaches all the tubes.

The boiler may also be cleaned by blowing the tubes from the front if preferred.

When using superheated steam of high temperatures it is necessary to provide some lubrication for the pistons and valves in the main engine. This has been the source of much discussion. The fear of using oil in the steam for marine work has probably delayed the installation of superheaters in steamers, but present experience indicates that with the proper use of high-heat mineral oil, and with efficient oil extractors, there need be no trouble from oil in the boilers.

There are several types of oil filters which may be used for this service, some of which consist of a simple form of a cartridge in which the water enters in the middle and is forced through an annular space filled with cocoa fiber; this filter may be fitted both on the suction side and the discharge side of the pump. Some engineers provide for large filter tanks and slow movement of the water to enable any oil to be collected.

ARRANGEMENT OF FIREROOM.

In large passenger vessels the arrangement of the boilers and firerooms to provide for efficient operation and the least interference and encroachment upon passenger spaces are points for special study by engineers and naval architects. The arrangements in the Hamburg-American Line steamers *Vaterland*, *Bismarck* and others, in which the uptakes are carried to the stacks from the outboard sides of the firerooms, thus permitting through fore-and-aft passages in the saloons and public rooms, are worthy of mention.

The arrangement of screen bulkheads in the firerooms providing large downtake spaces for fresh air, and enclosed spaces over the boilers, extending to the fronts of the uptakes, for carrying away the heated air, provides good stokehold ventilation for ordinary vessels and improves the fireroom conditions as frequently observed.

The demand in naval work to reduce the number of stacks and to armor the uptakes above the protective deck, in order to protect the boiler installation, requires long uptakes, and hence the importance is seen of maintaining the lowest temperatures in the uptakes under forced conditions.

The escapes from airtight firerooms should start from air locks at the bottom of the fireroom to minimize the danger in case of accidents which might fill the boiler room with steam.

The arrangement of side bunkers in large merchant vessels provides for a greater coaling length, which materially facilitates coaling in port.

BOILER-ROOM PIPING, VALVES, ETC.

The steam piping from the boilers should be arranged with drainage trapped or led back to the boilers and with sufficient freedom for expansion. There is a tendency to abandon copper material for steel, and with high temperatures and pressures the piping should be of seamless steel.

Automatic self-closing valves are sometimes installed as boiler stop valves to prevent, in the case of the bursting of a boiler tube, the steam and water from other boilers blowing through the disabled boiler, but such devices are not very popular. Engineers usually prefer to have the valves positively operated from place and from deck or from an adjacent compartment.

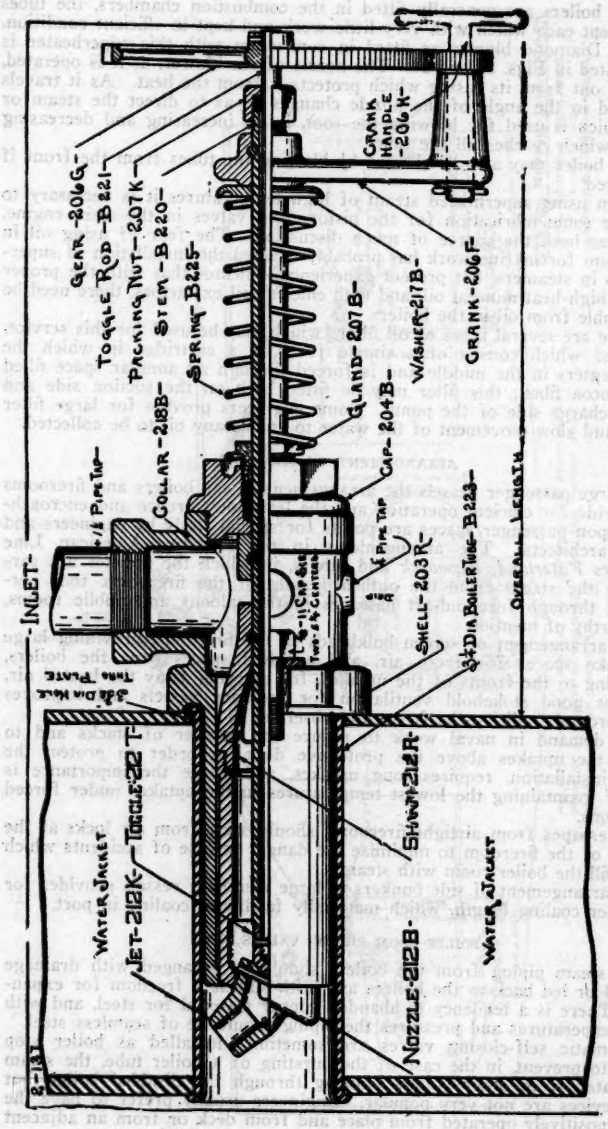


FIG. 25.—DIAMOND SOOT BLOWER.

Blow valves are now being made of the seatless hollow-piston type, with clear opening to give the least resistance and without projections upon which sediment may accumulate.

Safety-valve regulations for merchant work are prescribed by the various Governments and classification societies; these do not present the uniformity desirable. If rules could be adopted by the principal maritime countries prescribing the same essential characteristics for such valves, it would be of much advantage. Corresponding inconsistencies exist in the rules governing boiler scantlings.

Fig. 31 indicates how, for a special case of a Scotch boiler of given diameter, the allowed working pressure, or the thickness of the shell, varies under different classification societies and Government regulations.

The same variations apply to the steam and water drums of water-tube boilers.

BOILER-ROOM COMMUNICATION.

This subject is becoming more important owing to the increased size of boiler installations, the tendency to greater subdivision of boiler- and

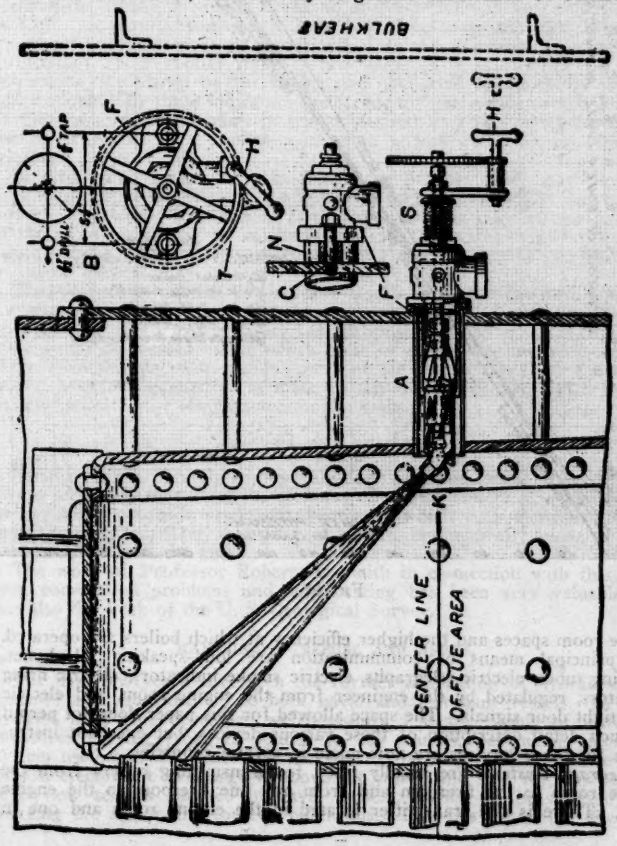


FIG. 26.—ARRANGEMENT OF DIAMOND SOOT BLOWER IN CONNECTION WITH SCHMIDT SUPERHEATERS.

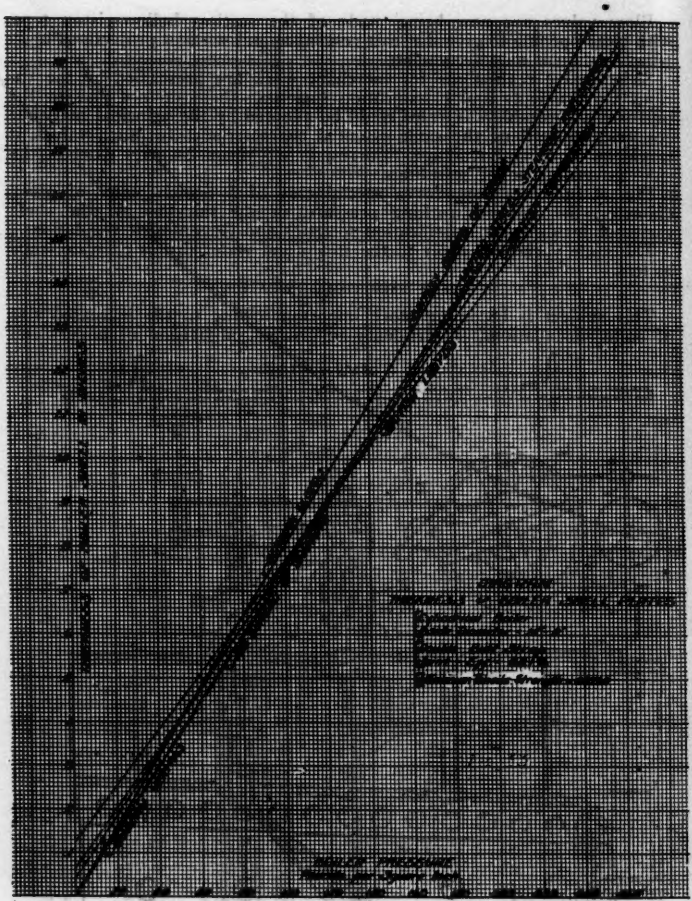


FIG. 27.

engine-room spaces and the higher efficiency at which boilers are operated. The principal means of communication are loud-speaking telephones, speaking tubes, electric telegraphs, electric smoke indicators, electric firing indicators, regulated by the engineer from the engine room, and electric watertight door signals. The space allowed for this paper does not permit of much detail description of these various devices, but excellent instruments for these services are now made in several countries.

Telegraph Systems are usually fitted for transmitting orders from the engine room to the fireroom and from any one fireroom to the engine room. There is one transmitter located in the engine room and one in

each fireroom, each instrument being a combined transmitter and indicator. The signals consist of lamps mounted in the instruments behind glass plates which cover orders or indications perforated in the face of the dial. Attention in each room is called to the visual signal by means of a gong which rings once for each signal given.

Firing Indicators for controlling the firing of boiler furnaces have been adopted in many naval and large merchant vessels, and it is only by such means in large installations that the maximum coal economy and minimum smoke production are realized. With this system the boiler furnace doors are numbered and an indicator provided in each stokehold signals these numbers in rotation at regular intervals, at the same time sounding a gong to call attention to each change of signal. The interval between successive signals is controlled by the regulator placed in one of the engine rooms.

Smoke Indicators.—The density of smoke is best defined by reference to the Ringelmann chart as published by the U. S. Geological Survey.

Smoke-indicator systems are sometimes installed where it is difficult to observe the smoke from the fireroom, and especially in connection with fuel oil. In naval vessels where it is desired to cover the maneuvers of the vessel under cover of a dense cloud of smoke, a means of giving this information to the boiler room is necessary. The transmitter for this system may be placed on the bridge and connected to indicators in the boiler rooms. By these indicators the freedom from smoke or the density of the smoke or the amount of smoke desired may be communicated to the boiler rooms in a similar manner to orders given on the electric telegraph.

In oil-burning boilers it is customary to provide small glass-covered openings in opposite sides of the uptake passage as the smoke leaves the boiler. An electric light placed at the back opening may be clearly seen at the front opening if no smoke is present. A mirror is placed at the front opening to reflect the image where it may be seen by the operator of the oil burners.

Electric pyrometers for uptakes and stacks may be mentioned here, as the temperatures in these places are so closely connected with the efficient operation of large marine-boiler installations. The dials of such pyrometers may be located in the boiler room or elsewhere, and wires led to thermo couples placed in various parts of the gas passages; by the turning of a switch, connection is made to any desired couple, which immediately indicates the temperature on the dial.

ANALYSIS OF FLUE GASES.

Facilities for accurately and quickly analyzing flue gases have been the means of largely increasing the economy of marine boilers. Several forms of apparatus are now used aboard ship by which such analyses can be made continuously and a permanent record automatically made of the amount of CO_2 in the gases.

The work of Professor Robert H. Smith in connection with this subject, combustion problems and superheating has been very valuable, as has also the work of the U. S. Geological Survey.

HANDLING OF REFUSE AND ASHES.

The past few years have seen the further application of hydraulic ash ejectors and ash expellers to both naval and merchant vessels. Several systems have been developed which require a steady supply of water at pressures varying from 15 pounds to 300 pounds, depending upon the system used, and the draft and the size of the ship. With the See type of ash ejector the discharge is above the water line, and for this reason it is now seldom adopted in naval vessels, owing to the necessity of piercing the armor and torpedo bulkheads. This has led to the develop-

ment of the hydraulic ash expeller, which employs the principle of an induced water current. The ashes are carried into the discharge pipe and forced through the bottom of the vessel or out through the side of the ship below the water line. There are automatic safety features embodied in most designs. In installing this type of ejector the discharge must be so placed and directed as to prevent ashes and cinders from being drawn into the sea suctions. Figs. 28, 29, 30 and 31 represent the Stone, the Metten and Palen-Burlingham types of these expellers.

Refuse hoists are usually fitted even with oil-burning installations, and ash hoists are also installed in conjunction with ash expellers. Such hoists consist of automatically-controlled double-cylinder engines, steam tubes or tubes operated by air pressure or by the vacuum system. Ash hoists are often arranged with self-dumping buckets which empty into chutes, where the ashes are flushed overboard. Some vessels have chain ash elevators which discharge to similar hoppers and chutes.

MECHANICAL AND NATURAL DRAFT.

The practice of fitting Howden draft in connection with Scotch boilers is still increasing. It is a tribute to the foresight of Mr. James Howden that the Howden draft installations of today are so similar in most essential points to the installations which he made fifteen or eighteen years ago.*

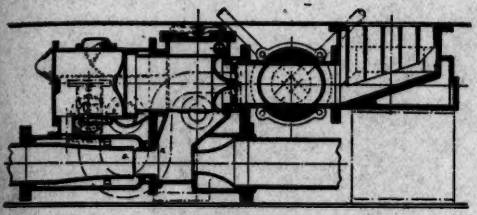
Howden draft is best adapted for use in connection with Scotch boilers, although some large installations of water-tube boilers have recently been made using this system of draft. The Howden principle of automatically closing all dampers to each furnace as fired makes the system safe and well adapted for Scotch boilers with separate combustion chambers. In applying this draft, or any similar system, to large, common, furnace water-tube boilers the air must be wholly or partly cut off when firing; thus a large grate area is affected, and for this reason, with such boilers, induced draft, which does not require closing of the draft dampers when firing, is sometimes more practicable.

A few years ago when induced draft with air-heater tubes was more commonly applied to Scotch boilers than at present, dampers in the uptakes for each furnace were arranged to be automatically closed when firing. The principal objections to induced draft are the increased size of fans, the difficulties of arranging the air ducts, and the fact that a large amount of cold air is allowed to enter the furnace when firing, due to the fan suction.

Some water-tube boilers are now designed with heating surface, gas passages and circulation so proportioned and arranged as to reduce the uptake temperatures sufficiently for good economy without the use of air-heater tubes.

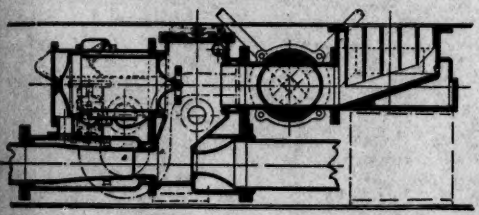
It has been suggested in connection with large electrical propelling machinery installations, in which it is necessary to ventilate the main generator and propelling motors, to adopt the arrangement of placing the main power-generating units amidships, with boilers located both forward and aft of these units, and with the propelling motors aft of the after boiler installation, so as to best utilize the warmed air from the ventilation ducts of the electrical machines in connection with the forced-draft system of the boilers. This also has the additional advantage of reducing the length of the steam piping to the turbo-generators, which is a decided advantage in large installations, owing to less drop in the steam pressure at the turbines.

* Mr. Howden in a late paper on "Forced Combustion in Steam Boilers," reviewed this subject and stated that "The honor of first using a blowing fan to accelerate combustion in a steamboat belongs to Mr. Edwin A. Stevens of Bordentown, New Jersey, who in 1827, in the *North America*, fitted boilers with closed ash pits into which the air of combustion was forced by a fan."



EXPELLER SHUT OFF.

Scale ½ Full Size.



EXPELLER WORKING

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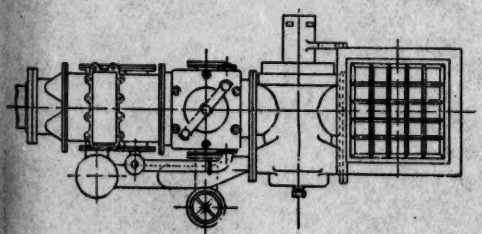
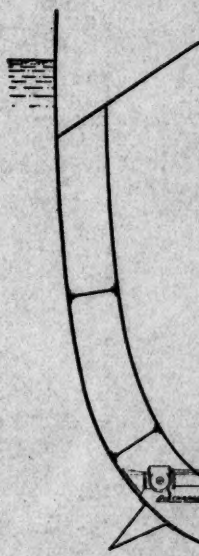


PLAN ON STOKEHOLD FLOOR

FOOT-PLATES REMOVED

Scale ½ Full Size.



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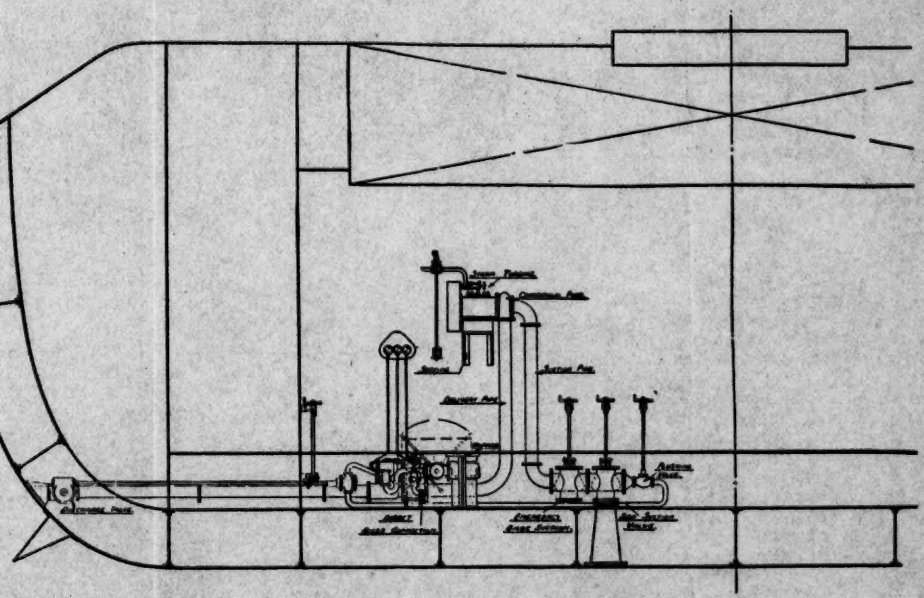
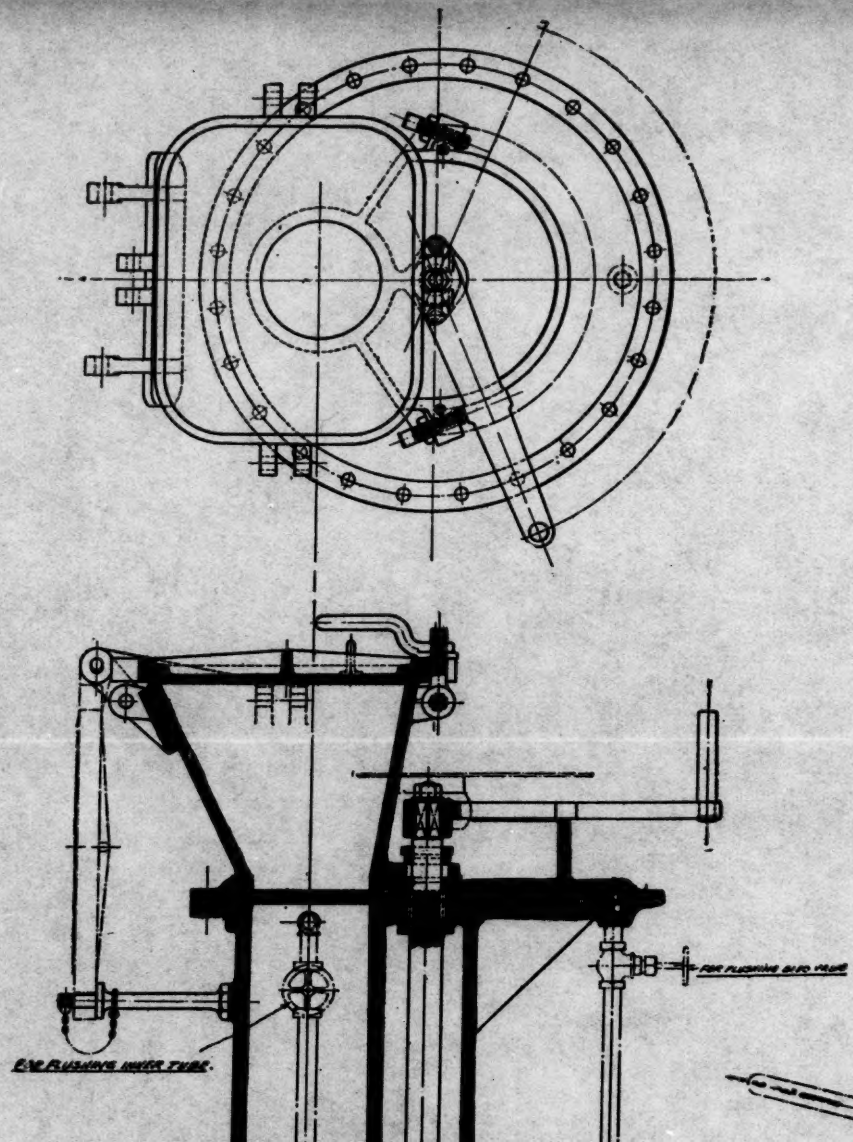
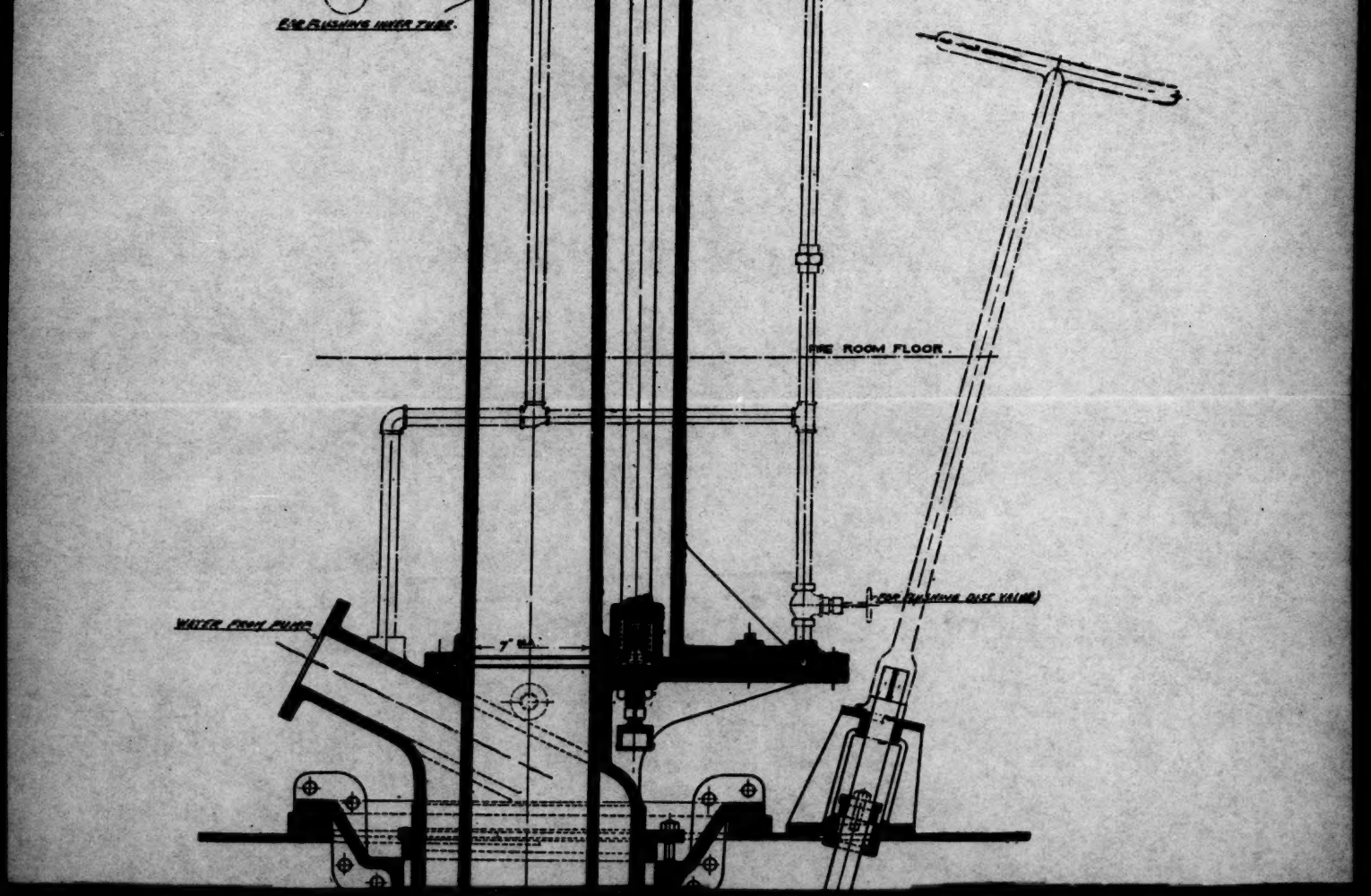


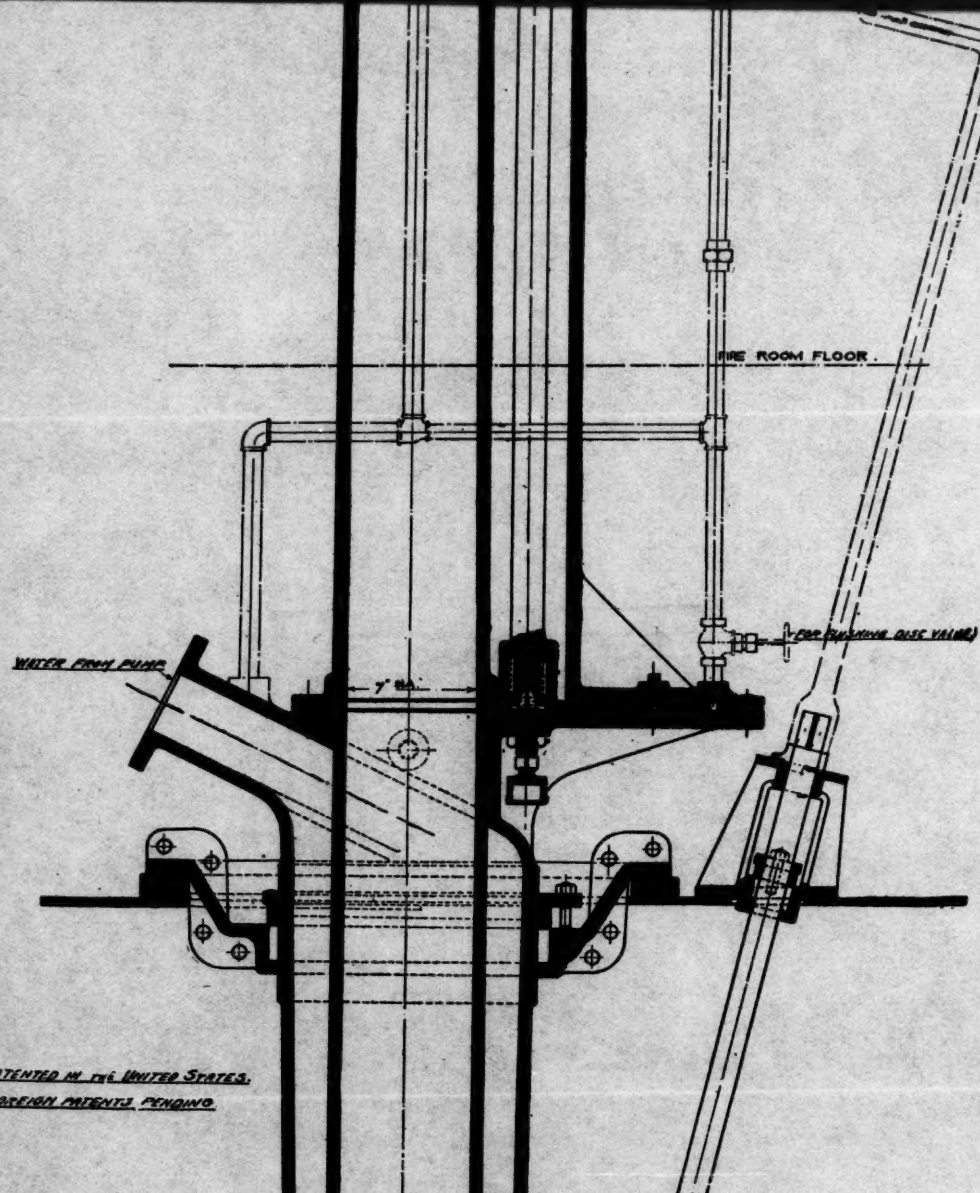
SECTION OF SHIP SHOWING GENERAL ARRANGEMENT OF STONE'S ASH EXPELLER.

NOTE:—THE DISCHARGE MAY BE LED OUT AT ANY POSITION FROM IMMEDIATELY BELOW THE WATER LINE TO THE KEEL AS MAY BE FOUND CONVENIENT IN RELATION TO OTHER FITTINGS.

FIG. 28.


FOR FLUSHING OUTER TUBE.
FOR FLUSHING OUTER TUBE


E.W. FLUSHING INNER TUBE.
ENGINE ROOM FLOOR.
WATER FROM PUMP.
7"
FOR FLUSHING DISC VALVE)



PATENTED IN THE UNITED STATES.
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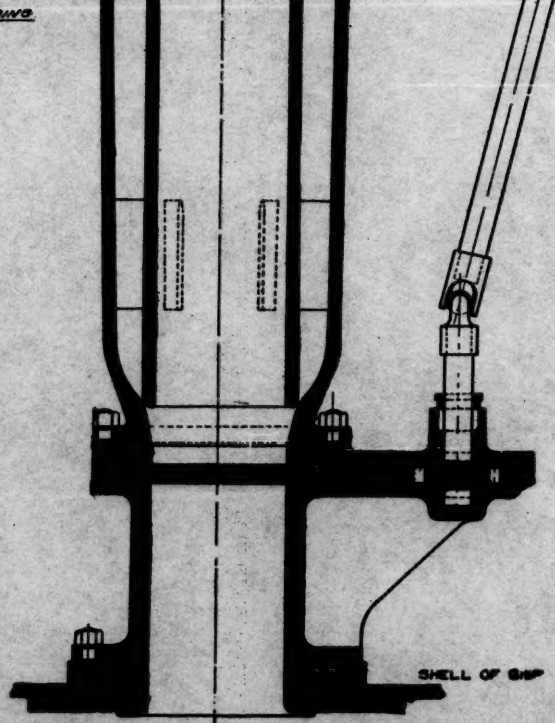

FIG. 29.—METTEN ASH EXPELLER—U. S. S. "CYCLOPS."

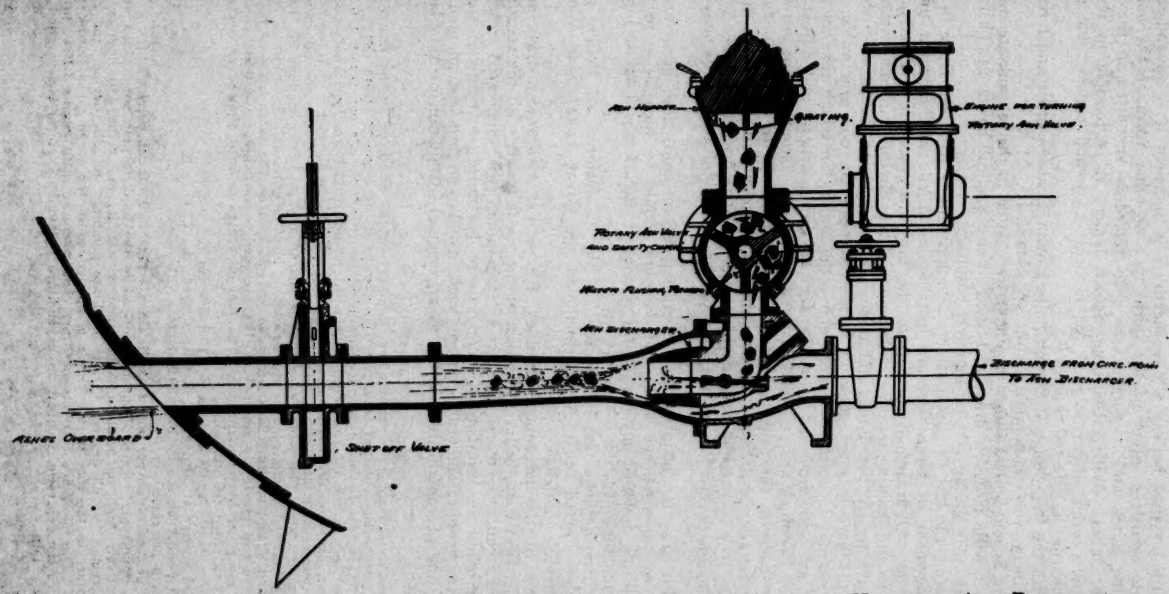

FIG. 30.—SECTION SHOWING OPERATION PALEN-BURLINGHAM UNDERWATER HORIZONTAL HYDRAULIC ASH DISCHARGER.

With fire-tube boilers, where the draft is sufficient it is advantageous to use retarders, either of the spiral type or other form, in the boiler-heating tubes, and in some cases such retarders are fitted in the air-heater tubes. With oil-burning Scotch boilers it is feasible to install retarders even when designed for natural draft, and considerable gain in efficiency is effected thereby.

With the best-designed Scotch boilers, with Howden draft and modern machinery, very economical results are obtained.

The Scotch boilers shown in Fig. 1 were fitted in the oil-tank steamship *Topila* with Howden draft and with White fuel-oil burners. This installation is giving very economical results.*

During the last decade considerable improvement has been made in the arrangement of large fireroom installations fitted with Howden draft. In some well-arranged installations the fans are placed in pockets adjacent to and opening into the stokeholds, so that the air for combustion is drawn down the ventilators and across the firerooms to the Howden draft fans. Where this arrangement is neither desirable nor practicable, the blowers may be located on deck above the boilers, in which case suction ducts are sometimes led from the fans to the stokehold, and adjustable suction openings are provided in the fan casings, communicating directly or by ducts to the fan rooms to insure ventilation. In smaller installations the fans are usually fitted in the engine room with discharge ducts leading to the heater boxes, in which case the suction is taken either from the engine room or from the fireroom or from both spaces, as may be desired.

With large fan installations the capacity of the units and the arrangement of fan-discharge ducts and dampers should be such as to permit the closing down of any blower for repairs or attention without seriously affecting the operation of the system.

Such an arrangement is indicated in Fig. 32, Diagram of Boilers and Howden Forced-Draft Arrangement on the Great Lakes Side Wheel Steamer *See-and-Bee*.

Some of the advantages claimed for heated mechanical draft are: Smaller first cost; better utilization of the waste gases, with the resulting greater economy; more satisfactory regulation, with increased reserve power when the installation is correctly proportioned; economy of space and less weight; and, furthermore, such installations are best adapted to the fitting of superheaters, either as an original installation or as a later change.

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* Mr. A. S. Hebble, Superintending Engineer of the Southern Pacific Company, states that he has obtained on a trip from New York to Tampico, using Navy fuel oil of 26 degrees Baumé gravity containing about 19,500 B.t.u.'s per pound (10,730 cal. per kg.), a fuel-oil consumption of 0.875 pounds (0.397 kgs.) per horsepower per hour, based on the I.H.P. of the main engines and an allowance of 7 per cent. for auxiliaries and for steam used in the oil heaters and in the cargo and fuel-oil tank heater coils. The same vessel for the last 15 voyages, prior to June 23, 1915, running between Tampico, Mexico, Galveston, New Orleans and Sabin Pass, New York and Tampico, using Mexican crude fuel oil, and on the same basis of power as on the trip previously referred to, has averaged a consumption of 0.981 pounds (0.445 kgs.) per horsepower hour. The fuel oil for these trips showed an average analysis of about the following:

Specific gravity at 60 degrees F. (15.6 degrees C.) 0.9799.
Baumé gravity at 60 degrees F. (15.6 degrees C.) 12.87.
Flash point, degrees F. 139 (59.5 degrees C.).
Water 1 per cent.
Heat of combustion per pound of oil, B.t.u.'s 17,976 (9,980 cal. per kg.).

It is important to keep Mexican crude oil in the bunkers and cargo tanks at a comparatively high temperature, approximately 100 degrees F. to 110 degrees F. (38 degrees C. to 43 degrees C.), to facilitate pumping and discharging, and this temperature in the case of the *Topila* is maintained in the cargo tanks during the voyage. The allowance therefore of 7 per cent. over the main engine I.H.P. probably does not represent as great an expenditure of auxiliary steam consumption as actually obtains.

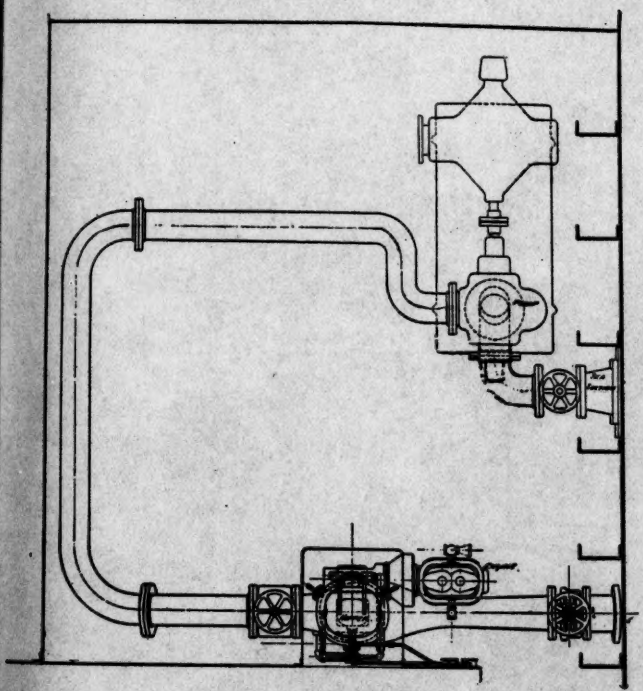
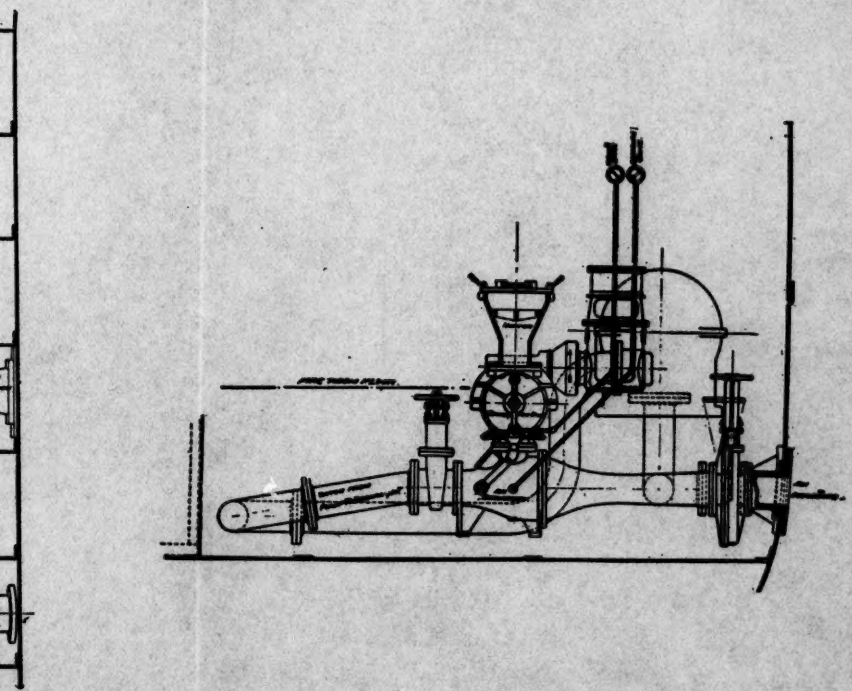
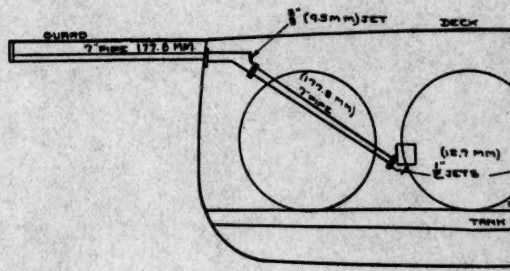


FIG. 31.—ARRANGEMENT OF PALEN-BURLINGHAM U



LINGHAM UNDERWATER HORIZONTAL HYDRAULIC ASH EXPELLER.



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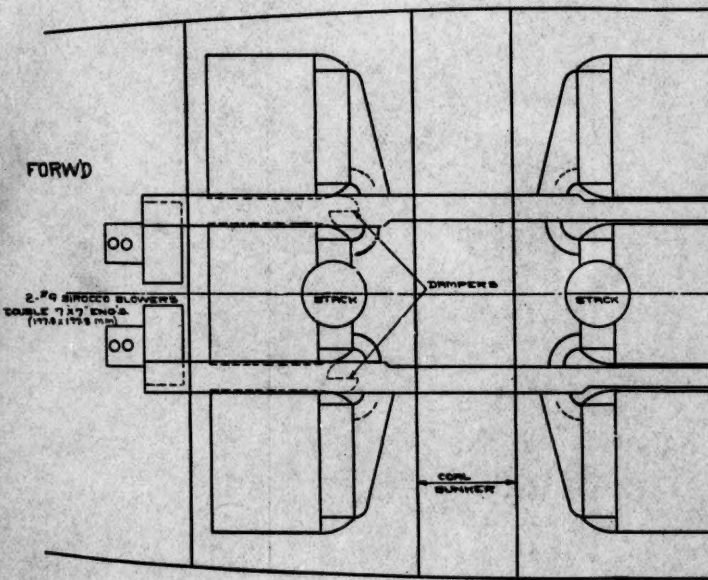

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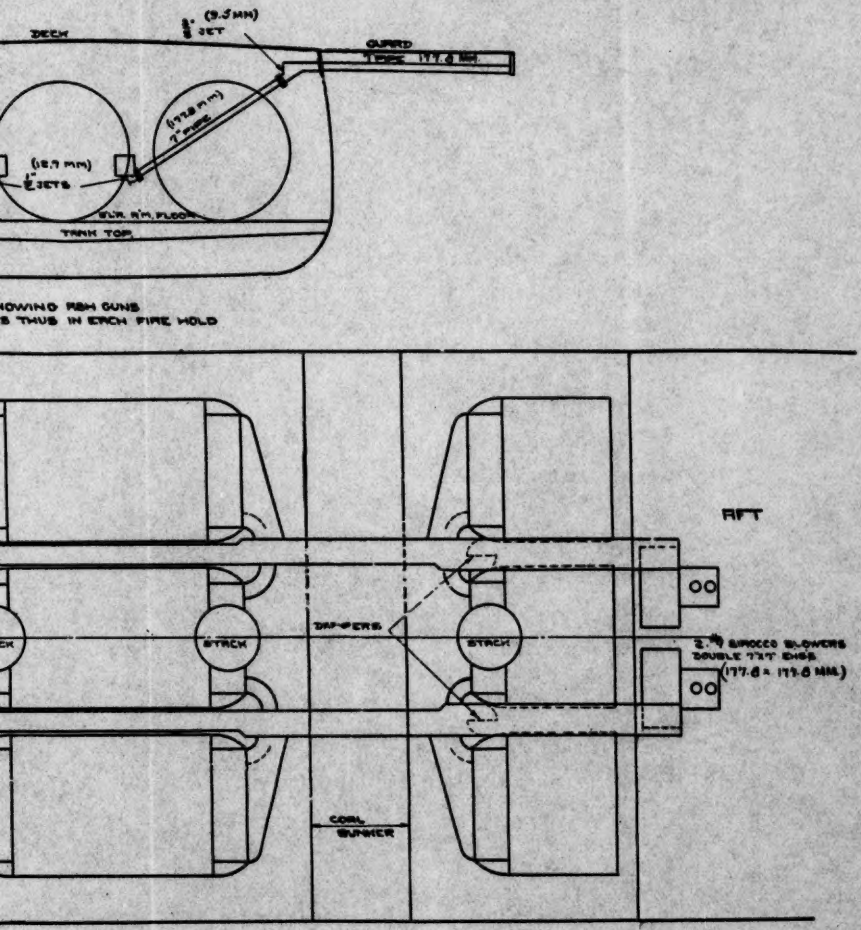


DIAGRAM OF BOILERS AND HOWDEN FORCED DRAFT ARRANGEMENT

GREAT LAKES STEAMER "SEE AND BEE."

FIG. 32.

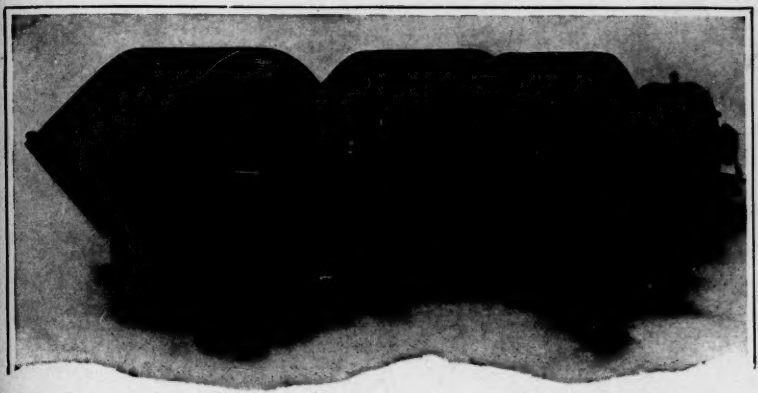


FIG. 33.—DOUBLE-INLET SIROCCO FORCED-DRAFT FANS DRIVEN BY DIRECT-CONNECTED STEAM TURBINE FOR TRAINING CRUISER FOR THE CHINESE COMMISSION. CAPACITY EACH FAN 30,000 CU. FT. PER MINUTE (850 CU. M.)3-INCH STATIC PRESSURE (76 M.M.) 1,220 REVOLUTIONS.

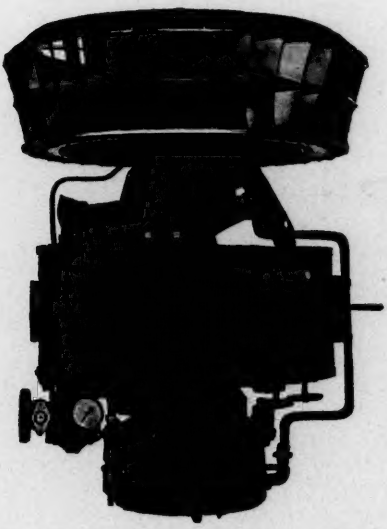


FIG. 34.—TERRY VERTICAL-SPINDLE TWO-BEARING RADIAL-FLOW FORCED-DRAFT FAN FOR ITALIAN AND U. S. DESTROYERS.

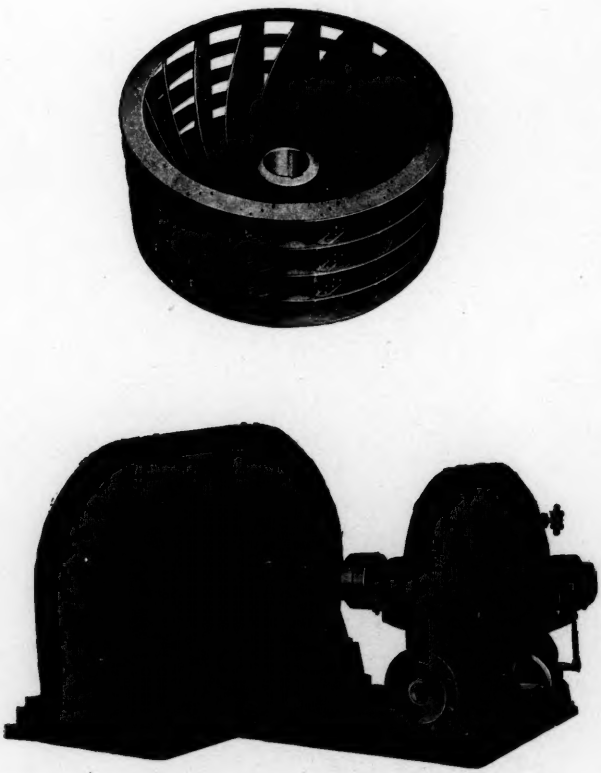


FIG. 35.—STURTEVANT HORIZONTAL TURBINE-DRIVEN FORCED-DRAFT FAN AND FAN WHEEL—BATTLESHIP "PENNSYLVANIA."

For several years past large installations of fans, either in connection with Howden draft or the closed-fireroom system or as fireroom ventilating fans for natural draft, have been made with high-speed fans of either the steel-plate type or the multi-vane type driven by electric motors or steam turbines. Such installations are sometimes fitted with the fans arranged in pairs with a steam turbine or a variable-speed shunt-wound self-ventilating motor between each pair. Water gages and tachometers are provided at the fans. If electric motors are used the controllers are arranged to be operated both at the fans and in the stokeholds and are made to protect the motors against over loads and failure of supply. The motors can be stopped by press buttons located in the fan rooms and the stokeholds.

Closed-fireroom installations are usual in naval work.

Fig. 33 shows "Sirocco" fans driven by turbines direct connected for forced draft, on a training cruiser for the Chinese Commission.

When two or more motor-driven fans of this character are installed for each closed fireroom it is advisable to fit an automatic damper at each fan discharge so balanced that if a circuit breaker for any fan motor blows, the air pressure in the fireroom will not reverse the motor and due to reduction in the air pressure throw a large excess load on the other fans.

The turbine-driven fan is probably the most compact and lightest outfit which has been put on the mraket for outfits requiring moderately-high air pressures.

Fig. 34 shows turbine-driven fans built by the Terry Steam Turbine Company designed for forced draft on Italian and U. S. destroyers. The casing is split vertically; a small portable crane is arranged to take the weight of the covers when dismantled. Each fan for the Italian boats is designed to deliver 34,000 cubic feet (963 cu. m.) of air per minute against an air pressure of 5.9 inches (150 mm.) water when running at 70 horsepower and at 1,400 revolutions per minute. These fan wheels are 37½ inches (952 mm.) diameter.

Turbine-driven fan units have been fitted in several destroyers arranged to work compounded, the high pressure exhausting at from 50 pounds to 60 pounds (3.4 to 4.1 atmos.) pressure, the low pressure exhausting at about 10 pounds (0.7 atmos.). This results in a saving of about 20 per cent. in the water rate as compared with single units.

Fig. 35 shows one of twelve Sturtevant horizontal steam turbine-driven forced-draft fans which are being fitted in the U. S. Battleship *Pennsylvania*. Each blower has a 25-inch (635 mm.) cone fan in a scroll casing with double inlets, and is direct connected by a flexible coupling to a Sturtevant steam turbine. Each unit will deliver 17,500 cubic feet (496 cu. m.) of air per minute at 4 inches (102 mm.) air pressure in the fireroom when making 2,100 revolutions per minute, and is able to maintain an excess capacity of 20 per cent. with steam pressure of 250 pounds (17 atmos.) with an exhaust pressure of 15 pounds (1 atmos.) gage pressure at the turbine.

Each turbine is fitted with an automatic governor and forced lubrication, which is closed to prevent dust from coming in contact with the oil.

The Keith fan has shown great efficiency and is well adapted to mechanical-draft purposes.

Only a few years ago when it was proposed to install steam turbine-driven fans for this class of work, strong criticisms were raised that such fans would be unsatisfactory. It is now demonstrated that these fans are efficient, safe, economical and reasonably quiet. Similar fans to those mentioned above are now made by the principal fan builders, who deserve much credit for bringing these units to the degree of efficiency and dependability which has been realized.

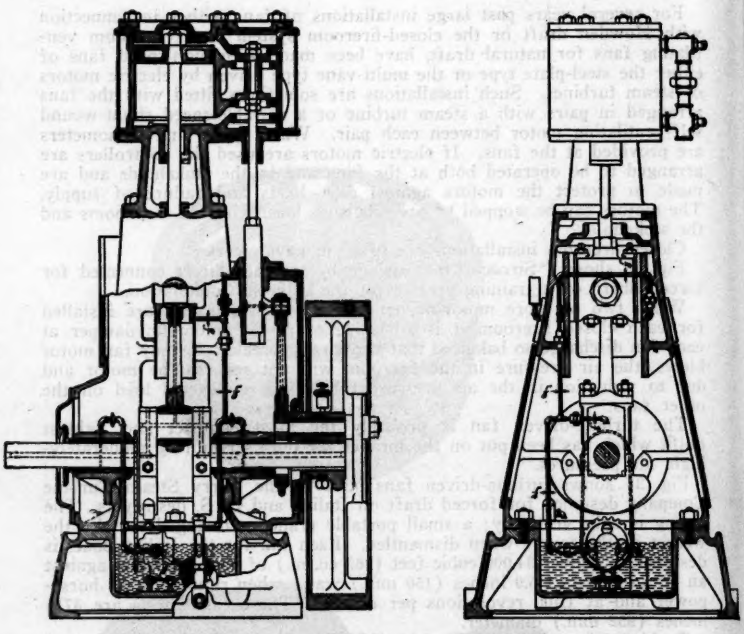


FIG. 36.—STURTEVANT FAN ENGINE, FORCED LUBRICATION.

When two or more turbine-driven high-speed fans are installed in a single fireroom, the speed of each fan must be practically the same to obviate the fans working against each other. The fan speeds may be adjusted by a single valve at each turbine and all fans slowed down or speeded up, as conditions require, by the manipulation of one valve supplying steam to all the turbines.

Some steamship lines favor natural-draft boilers for their vessels, but in large installations provide forced fan ventilation to the firerooms. Rocking grates with natural draft are also fitted as an assistance in working the fires. Mechanical stokers have been tried in several installations, but the problems of their adoption in marine work have not yet been solved.

BOILER FEED PUMPS.

The tendency in steam feed-pump design has been to eliminate flat surfaces, substituting spherical and cylindrical parts to stand the high pressures and still maintain as light weights as feasible.

The long-stroke vertical simplex boiler-feed pump has gradually developed towards a better steam economy, a more positive operation and greater ease of overhauling.

The high boiler pressure of 250 to 295 pounds (17 to 20 atmos.) and the use of superheat have required a valve gear that will operate without lubrication.

The Blake & Knowles Company have developed a design of suction and

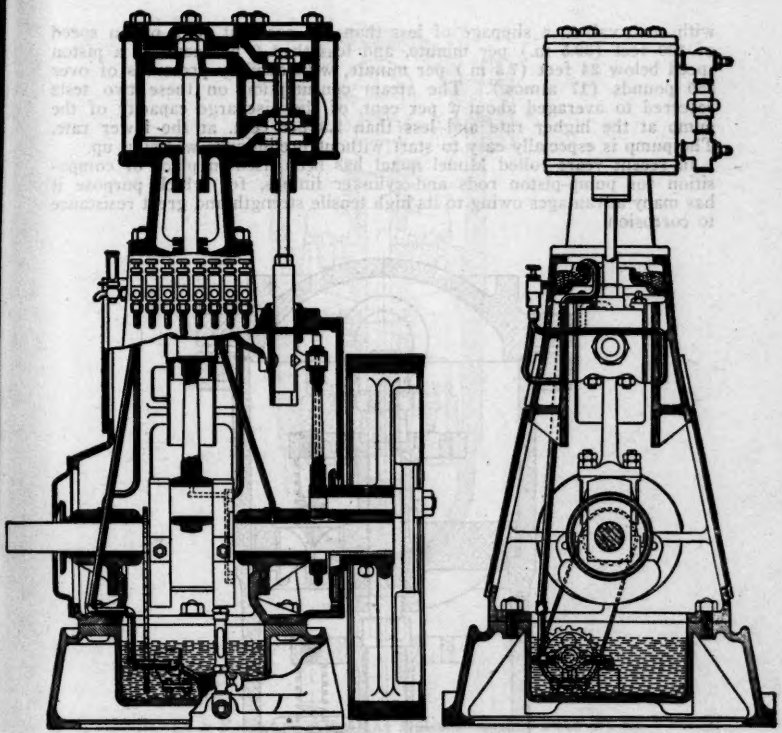


FIG. 37.—STURTEVANT FAN ENGINE, AUTOMATIC GRAVITY LUBRICATION.

discharge valves, shown in Fig. 38, which permits of the withdrawal of the entire set of valves by the removal of only the covers and lock nuts.

Bevel-faced wing-guided valves are sometimes used and also valves of the three-disc type with suitable valve decks.

Another design, Fig. 39, provides annular double ports which permits of a very low lift of valve and a minimum slip and noise, facilitating high speed.

Rubber valves used in connection with cold-water pumps are ordinarily subject to considerable wear. This difficulty has been practically eliminated by the development of the rotating valve. The feature of this design is the partial rotation of the valve at each stroke as it is lifted from its seat, which causes the contact between the valve and the seat ribs to change continuously, thus distributing the wear over the entire face with the result of greatly increasing the service of the valve. This rotation is secured by inclining the ribs of the valve seat so that the water in passing impinges on the valve at an angle.

Fig. 40 shows a feed pump of the simplex vertical type built by the Warren Pump Company. This design likewise incorporates the triplex suction and discharge valves. On test* this pump showed, when fitted

* Test of a Warren vertical single boiler-feed pump, Naval Engineering Experimental Station, Annapolis, "Journal A. S. N. E.," May, 1913, Vol. XXV.

with such valves, a slippage of less than 1½ per cent. at a piston speed of 100 feet (30.5 m.) per minute, and less than 9 per cent. at a piston speed below 24 feet (7.3 m.) per minute, with delivery pressures of over 250 pounds (17 atmos.). The steam consumption on these two tests referred to averaged about 2 per cent. of the discharge capacity of the pump at the higher rate and less than 2.8 per cent. at the lower rate. The pump is especially easy to start without preliminary warming up.

In recent years rolled Monel metal has been used in place of composition for pump-piston rods and cylinder linings, for which purpose it has many advantages owing to its high tensile strength and great resistance to corrosion.

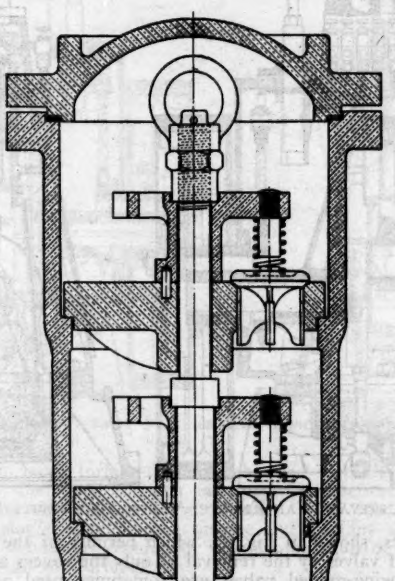


FIG. 38.—BLAKE & KNOWLES FEED-PUMP VALVE ARRANGEMENT.

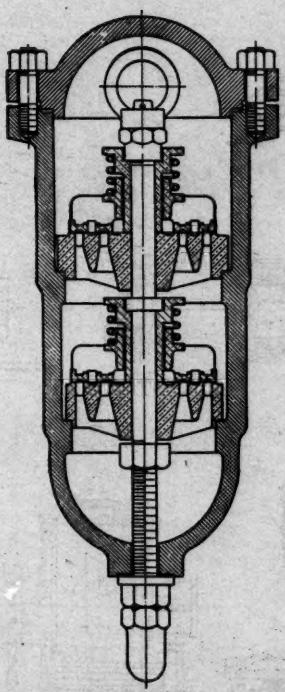


FIG. 39.—BLAKE & KNOWLES ANNULAR DOUBLE-PORTED FEED-PUMP VALVES.

Direct-acting pumps are now being built with a cut-off arrangement on the steam valves. This operates on the back of the main slide valve, which gives an improved steam economy and smooth, quiet action, with quick reversal but with a minimum shock in the pipe lines. Direct-acting pumps are well adapted to automatic control either by the discharge pressure or by feed-heater or tank-float control.

Centrifugal boiler-feed pumps have been adopted to a comparatively small extent, in some cases for boiler pressures as high as 295 pounds per square inch (20 atmos.). The continuous flow largely eliminates the shock through piping, valves and checks, although this can also be accomplished by the proper use of air chambers in connection with direct-acting long-stroke pumps.

As Mr. Weir has pointed out,* the feed pumps should be driven by the same source of power that actuates the main machinery. It is to be noted in this connection that turbines are being successfully developed for such pumps in a considerable amount of high-grade merchant and naval work. The feeling of caution as to the use of centrifugal pumps is changing to one of more assurance and confidence.

* "Development in Auxiliary Units Between Exhaust Pipe and Boiler," "Transactions of Institution of Engineers and Shipbuilders in Scotland," October, 1912.

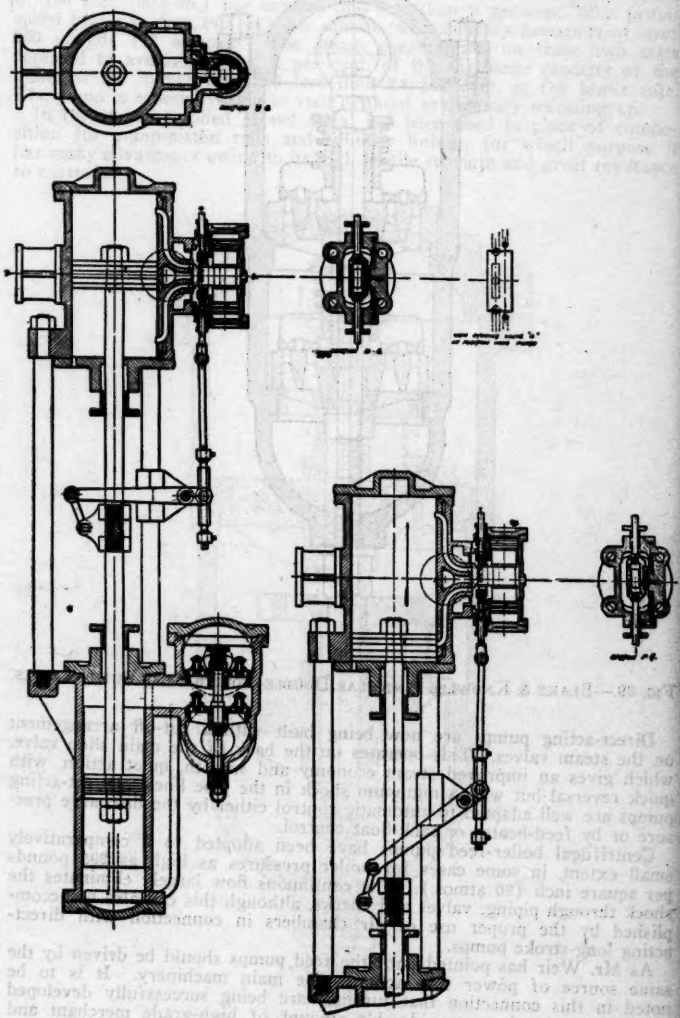


FIG. 40.—DETAILS OF WARREN PUMP.

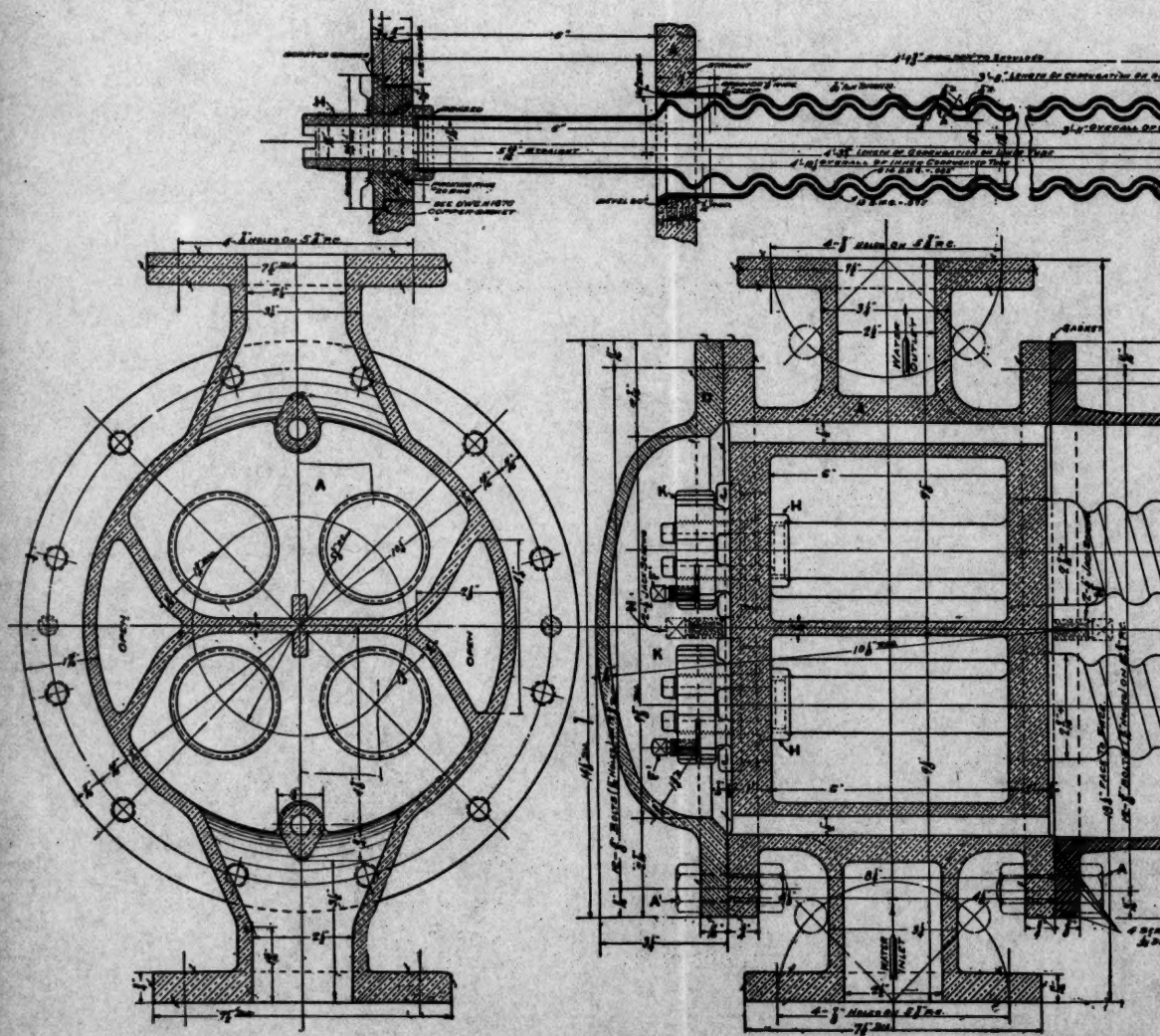
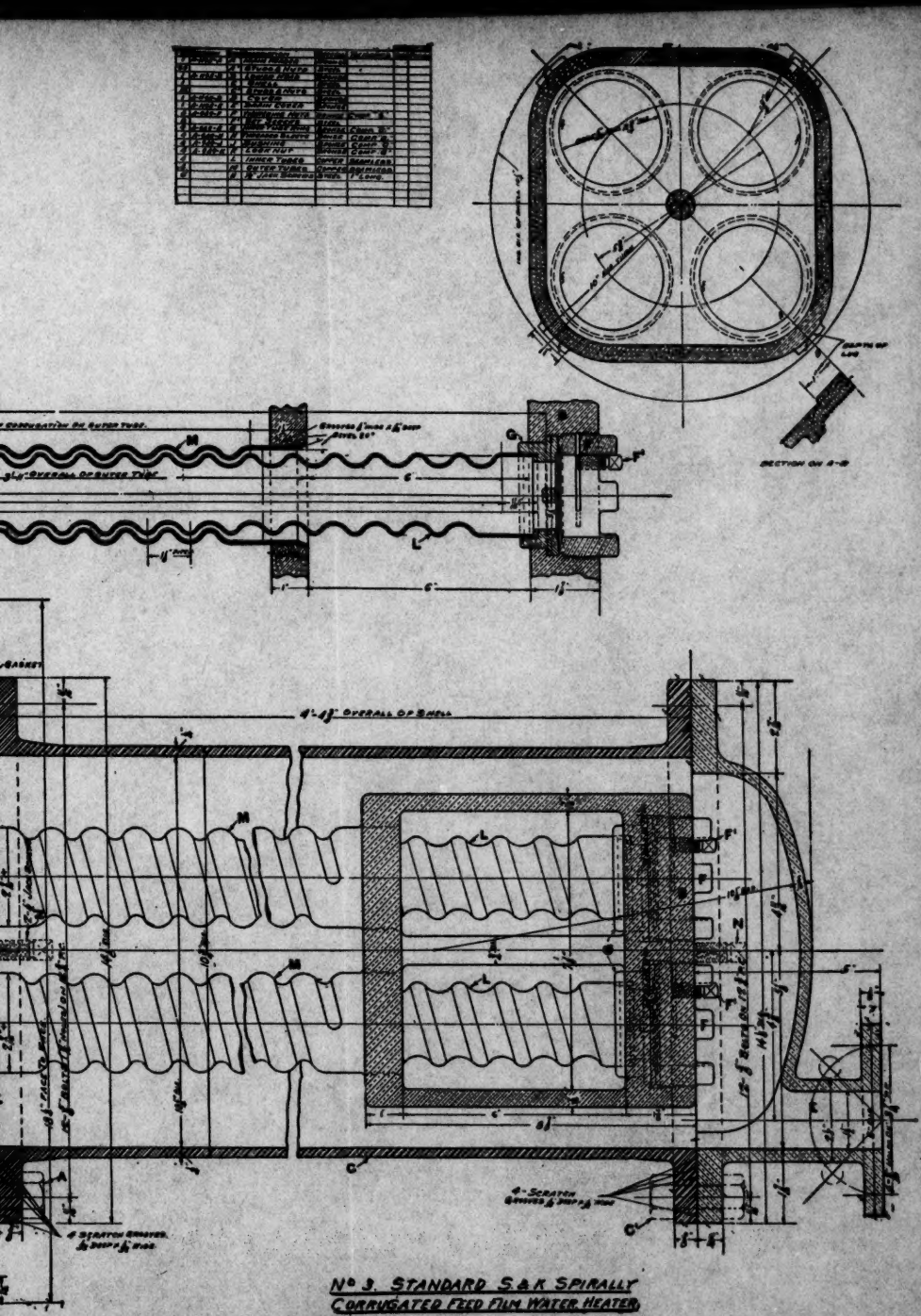


FIG. 41.—No. 3 STANDARD SCHUTTE & KOERTING SPIRALLY



SPIRALLY CORRUGATED FEED FILM HEATER.

FEED-WATER HEATERS.

The importance of feed heating is being more generally recognized in recent years. This is largely due to the fact that the amount of auxiliary exhaust in present installations is greater, owing to an increased number of independent auxiliaries.

In boilers where the feed is taken in so as to extract the greatest amount of heat from the escaping gases, feed heating is not so essential; but in most types of boilers it is important to fit auxiliary-exhaust feed heaters, both from the point of economy of fuel and boiler maintenance.

Steam feed-water heaters are fitted of two principal classes—the contact type, which is found in many trans-ocean liners, and the surface or pressure type, which is used in both naval and merchant work. The contact type makes it necessary to employ an additional pump for lifting the water from the hot-well to the heater and also entails handling the heated feed in the main feed pumps. This in some cases has proven unsatisfactory, but the ability of this heater to eliminate injurious gases from the feed and the high efficiency are points in its favor, and large numbers of this class of heater, including the Weir and Blake types, have been fitted.

If oil is used so that there is any possibility of its entering the steam or exhaust, special precaution must be taken to eliminate it. This heater also requires to be placed so high that it usually cannot be adopted for naval work. This point has been brought out and the subject of feed heating ably discussed by Mr. Weir in his paper above referred to.

The surface or closed type of heater is also used in merchant work and extensively in naval work. In some instances it may be advantageous to install both types of heaters in conjunction, as outlined by Mr. Weir. The surface type permits of higher feed temperatures than the contact heater.

The conditions governing the efficiency of surface heaters are lucidly presented by Prof. Leo Loeb.* After discussing the thermal resistances of several metals, water and air, Prof. Loeb states: "It is evident, therefore, that a metal tube will transmit all the heat that is presented to its surface and that the controlling resistances lie in the two films which cling to the metal surface. The condensing steam presents a wet surface, so that resistances on the two sides are much alike.

"The formation of such a film is a friction effect. The microscopic irregularities of surface of the metal walls tear off particles of the fluids passing and prevent these particles from being swept along with the major current. The more completely these particles of water are dislodged and swept from the surface, the better able are other colder particles to replace them and absorb their share of heat from the surface. Hence the problem in producing high transmissions in heaters is the destruction of this water film by a scrubbing action produced by a high velocity along the heating surface. When the limit of heat transmission has been reached on the water side by a velocity that is entirely practical, the controlling resistance has passed to the steam side, and the only way to further increase heat transmission is to sweep away the film on the steam side."

Prof. Loeb clearly outlines other conditions encountered in this type of heater and discusses heat transference as affected by temperature differences, the high transfer rates which may be obtained with water agitation, and the deleterious effect of air in reducing the efficiency of such heaters.

The experiments made by Prof. Loeb show the high efficiency possible with the Schütte & Koerting film type of heater. This heater is illustrated in Fig. 41.

* "Heat Transmission and Tube Length in Marine Feed-Water Heaters," "Journal of the A. S. N. E.," May, 1915.

BOILER CORROSION.

The corrosion of boilers has been the subject of much study and discussion. The principal causes of corrosion are the presence in the water of air and salts; acidity of the water due to salts and the use of animal or vegetable oils; and galvanic action. Corrosion is greatly increased by sluggish circulation. The feed water absorbs air and with it probably a greater percentage of oxygen than of nitrogen; salt water also absorbs more air than fresh water. With sluggish circulation the small bubbles of air cling to the metallic surfaces and attack the metal, especially in places where these bubbles are not continuously swept off by the water currents. Rapid circulation also retards scale formation.

Corrosion may be minimized by freeing the boiler of air, and therefore it is especially important to stop all air leaks in pipe connections, condensers, etc. Heating the water tends to drive out the air, and this is one of the advantages of the use of feed heaters which heat the water before it enters the boilers. It is also important to use pure water and avoid all water leaks in condensers which will tend to salt up the feed. It is probably advantageous to admit the feed to the steam space or slightly below the liberating surface of the water in the boiler to allow the air contained in the feed to separate.

Numerous boiler compounds have been marketed to remedy the troubles of boiler corrosion and scaling, but these should be used with caution, as some compounds are probably responsible for more serious and dangerous troubles. Zinc plates secured in the boiler are a preventive of corrosion, as is also the use of lime and soda as conditions require.

Some progressive boiler manufacturers issue instructions relative to the causes and the preventive steps to be taken and remedies to be applied in connection with corrosion and, in addition, furnish with each marine-boiler installation a testing kit to enable the engineer to ascertain the condition of his boiler feed and to take steps to minimize the troubles of corrosion and scale formation.

CONCLUSION.

In the last decade no revolutionary inventions have been made in this field, but several important new ideas are now being developed; no marked general increase in boiler pressure has been adopted; increase in size of units is conspicuous, as are very large power installations. Scotch boilers are slightly simplified and more dependable; great advance in reliability, efficiency, capacity and durability is noted in water-tube boilers; the use of such boilers in merchant vessels is increasing, which permits a large saving in weight and space. Howden draft for merchant work is still favored. Important improvements have been made in feed heaters, feed pumps, and forced-draft fans of steam and electric drive. The use of superheated steam is largely increasing and will probably continue, due to improved efficiency possibilities. Boiler-room communication, flue-gas analysis and ash and refuse handling have been largely perfected; boiler corrosion and boiler circulation are better understood and controlled; important advances have been made in methods of handling and burning oil.

A large opportunity is open to the various governments and classification societies in connection with standardizing rules and requirements governing the construction and use of marine boilers and boiler room equipment.

The writer wishes to express his grateful acknowledgments to the many friends in the engineering field, both in the United States and other countries, who have so materially assisted him with information, suggestions, data and drawings used in the preparation of this paper.

OIL ENGINES.

By G. E. Windeler.

An oil engine may be defined as an internal-combustion prime mover, which develops power by the introduction of liquid fuel or hydro-carbons to the combustion chamber of the power-producing or working cylinder. There exists an impression that the oil or liquid-fuel engine is a comparatively new form of power producer, having a limited sphere of operation, but we are informed that the first successful oil engine was set to work soon after the introduction of the gas engine. Its development was for quite a considerable period retarded, because it was thought impossible to use any but the most volatile oils, or oils having a low flash point.

Within the past decade its development has been such as to place it in a position of considerable importance as an economical and reliable prime mover, and the author feels that he is not presumptuous in claiming that it covers a broader sphere of utility as a power producer than any other engine in use at the present time. Most of the oil used is petroleum or rock oil, and distillates and residuums of petroleum. Petroleum is a mineral product, and is obtained from the earth in two different ways: (1) In a natural liquid state from subterranean wells of varying depth; (2) By distillation from bituminous shale. The distillation or extraction of oil from shale has been carried on in Scotland since 1850. The discovery of petroleum in a natural state followed several years later.

Petroleum, its distillates and residuums, have now become most important articles of commerce and industrial utility. The two most important oil-bearing countries are the United States of America and Russia, where vast interests are concerned in the collecting and distributing of millions of gallons per day. Oil is also found and commerced in very large quantities from Scotland, Mexico, Burmah, Galicia, also considerable quantities are found in Canada, Persia, Japan, China, Argentine; in fact, scientific men have expressed the opinion that oil may be discovered almost anywhere, if deep enough borings were made.

Petroleum in its virgin state is rarely used as a fuel, as it contains valuable by-products, such as petrol or gasolene, or benzine, kerosene, paraffin or lamp oil, paraffin wax and lubricating oils, which by-products are extracted from it by distillation at different temperatures. The distillates of petroleum, which are commonly used for fuels in internal-combustion engines, are petrol or benzine, paraffin or lamp oil, or kerosene or residuum.

Within recent years scientific research has placed within our reach other important fuels which are products of coal, and lignites, vegetable matter and animal fats, such as:

Benzol or Benzine,	Lignite Tar,	Alcohol,
Coke Oven Tar,	Lignite Tar Oils,	Palm Oil,
Vertical Retort Tar,	Earth Nut Oil,	Whale Oil.
Tar Oils,	Cotton-seed Oil,	

Natural selections have divided fuel oils and the engines capable of using them, generally speaking, into three clases: (1) Light volatile oils, such as petrol, benzine, benzol, as used by engines fitted with carburetors; (2) Medium flash point oils, such as lamp oil, paraffin or kerosene and solar oils, are used by engines fitted with vaporizers or hot bulbs; (3) High flash point oils, such as petroleum residuums, coke oven and vertical retort tars, tar oils, vegetable and animal fat oils, by engines fitted with pulverizing or atomizing devices.

Engines under Class 1 are commonly known as petrol engines, and are the most extensively used of all internal-combustion prime movers. These engines are operated on the two and four-stroke cycles, the four-stroke

cycle being the most common, although a fair number of two-stroke engines are used for small powers, such as required for motor cycles and boats. On the continent a large number of these engines use alcohol as fuel, but the heavy duties imposed on all forms of alcohols preclude its use for this purpose in this country.

The four-stroke cycle engine is usually fitted with two valves, an air inlet valve and an exhaust valve, both mechanically operated, and closed by spring pressure, and the cycle of operations is as follows: The piston on its down stroke draws atmospheric air through the inlet valve, which air, on its way to the cylinder, passes over a jet or jets in the carburetor.

This simple device consists of a small chamber connected to the supply tank; contained in this chamber is a float to which is attached a needle valve, which maintains the level of petrol so that it is just below the mouth of jet. The air passing over the jet at high velocity induces therefrom a spray, a petrol (which readily volatilizes at atmospheric temperature), thereby carburetting the air passing into the cylinder. At, or about, the end of the down stroke the inlet valve closes, and the mixture of air and petrol vapor is compressed for a portion of the up stroke of the piston. At a predetermined point it is fired, usually by an electric spark, and combustion and expansion takes place on the next down stroke; the products of combustion are expelled through the exhaust valve on the fourth stroke, completing the cycle. A power stroke is obtained every two revolutions.

Two-stroke petrol engines are usually arranged without air-inlet and exhaust valves, the piston itself acting as a valve by covering and uncovering ports which admit air and allow the exhaust gases to be expelled to the atmosphere. The ports are located at the lower end of the cylinder, and are arranged diametrically opposite to each other. As the complete cycle of filling the cylinder with air, compressing it, obtaining power, and exhausting, must be carried out every two strokes, expelling the exhaust gases and filling cylinder with air have to be carried out practically simultaneously.

To carry this into effect, air compressed above atmospheric pressure is supplied and is used to help to expel the exhaust gases and refill the cylinder with fresh air at the same time. This is carried out in a very simple and ingenious manner. As the piston ascends on the up stroke, air is drawn into the crankchamber (the engine being usually of the enclosed type) and is compressed by the displacement of the piston on the next down stroke, and passes from the crankchamber through a passage which is connected to the inlet port in the cylinder. Directly the piston on its next up stroke covers the ports, compression take place, then ignition, etc. A power stroke is obtained each revolution.

Owing to the volatile character of the oil, only a limited compression pressure is possible, with the result that a limited range of temperature is available, preventing the high economy in fuel consumption obtaining with engines using high compression; but improvement in the design of the engines and carburetors has resulted in very excellent fuel consumption now being obtained, *i. e.*, 0.55 of a pint of petrol with the four-stroke engine, and about 0.6 of a pint for the two-stroke engine, per B.H.P. per hour.

These engines are extensively used for small electric-lighting and pumping plants for country houses, for agricultural work, mechanically-propelled pleasure and commercial vehicles, also marine propulsion work for small pleasure and fishing craft. They have also been extenstively used for light fast craft, such as hydroplanes, patrol boats, racing boats, yachts, etc.

The early Holland submersible boats were fitted with this class of engine, but the appalling catastrophes that occurred very quickly brought about a decision that engines using such a volatile oil, which readily vaporizes at atmospheric temperatures were unsuitable when working in

confined spaces. Leakages were bound to take place, resulting in explosions and serious loss of life. Petrol engines have been satisfactorily produced developing powers as high as 600 B.H.P. It has also been largely responsible for the success of that beautiful but terrible engine of war, the flying machine, the latest wonderful production of man's genius.

It is generally conceded that the petrol engine is one of the most beautifully working prime movers produced. Its silence and smoothness of running, reliability, ease of starting and handling, flexibility and simplicity of construction, coupled with its extremely light weight per horsepower developed, has caused it to become a most popular source of power for the purposes named above.

Turning now to engines fitted with vaporizers, and using refined oil such as paraffin and the lighter crude oils, we find that Brayton, in the year 1873, built an engine which used lamp oil having a specific gravity of 0.85. Brayton holds priority in the application of introducing the fuel into the cylinder by means of compressed air. The consumption of fuel per B.H.P. hour of his earliest engine was 2.7 pounds. Experiments were carried out on this engine in this country by Messrs. Greenwood and Batley, of Leeds, but few, if any, of these engines were placed upon the market.

About the year 1888 Messrs. Priestman, of Hull, introduced an engine capable of using paraffin as fuel, and a considerable number were set to work, with most satisfactory results. They were largely used for agricultural work, for mine work, pumping, hauling and wood sawing, driving fog-signalling compressing apparatus, small electric-lighting plants and rock drills. It is interesting to note that one of the earliest of these engines was used for colliery work, being placed at the bottom of a winding shaft 1,650 feet deep, and employed for hauling tubs of coal up an incline of 500 yards long, gradient 1 in 10, and gave every satisfaction. The earliest engine gave fuel consumption of 1.73 pounds per B.H.P. hour. A 5 B.H.P. engine tested by Professor Unwin in 1889 gave an average consumption of 0.95 pounds per B.H.P. hour at full load, and 1.38 pounds at half load.

The engine was also adapted for marine work, in the vertical form, and one of the earliest applications was for the Manchester Ship Canal Co. Two barges for use on the Bridgewater Canal were each fitted with 10 B.H.P. engines, the load carried was 20 tons, the distance 17 miles, and the time taken 5¼ hours; a clutch-reversing gear was fitted. The Priestman oil engine was operated on the four-stroke cycle. Oil was delivered by a pump to a novel design of spraymaker, and was blown into a heated vaporizing chamber by means of compressed air at about 8 pounds per square inch, causing the oil to be instantly vaporized by contact with the heated surfaces of the chamber.

On the suction stroke air was drawn into the cylinder and with it the vaporized mixture, and on the next stroke the air and vapor were compressed to about 25 pounds per square inch when they were fired by an electric spark, supplied from a battery.

The exhaust gases were expelled through a mechanically-operated exhaust valve, and passed on their way to the atmosphere through a jacket space formed round the vaporizing chamber, and which by this means was automatically kept at a sufficient temperature to vaporize the oil. On starting, a lamp was necessary to preheat the vaporizing chamber, which operation took about twenty minutes.

Marine propulsion for small craft opened an extensive field for the refined-oil engine, and many firms make a specialty of engines designed expressly for this purpose. Generally speaking, the vertical engine is the type adopted.

Messrs. Gardner, of Patricroft, one of the leading firms who have made a careful study of the special requirements for marine propulsion, succeeded in producing a thoroughly simple, reliable, and economical high-

speed enclosed engine. The vaporizer of this engine is heated by a perfected burner, using the same fuel as the engine, and burning with a small, silent blue flame, the burner being incorporated with the vaporizer casing, in a compact manner, and encased in draft screens. The air is drawn into the cylinder through a jacket formed round the vaporizer, and the fuel, accurately measured to suit the load conditions, is delivered to the vaporizer by means of an ingeniously designed feed every time the air-inlet valve is opened.

Ignition is effected by electric spark: the air for the combustion being drawn through a jacket space surrounding the lamp-heated vaporizer keeps the vaporizer at a more constant temperature, and it is claimed that complete combustion is obtained with a comparatively lower fuel consumption, including the oil used by the lamp, than is the case with the exhaust-heated vaporizer. The regularity and smoothness of running are remarkably good, and a wide range of speed control is obtainable—an essential feature of marine propulsion engines.

Another feature in the design of this engine is the attention given to facilities for cleaning out the water spaces—an absolutely necessary feature, and one that is often overlooked.

Small craft are more often than not cruising about in muddy waters of harbors and creeks, and consequently their jacket spaces are liable to choke up with mud and other solid matter. A considerable number of these engines have been used on fishing trawlers and such craft.

Passing to the engines capable of using heavy crude oils, tar, tar oils, etc., this is a field in which the Diesel engine practically stands alone.

Whilst other types of high-compression engines can use heavy fuel oil for short periods, they must and do have considerable troubles, which are inherent to the principle upon which they work; whereas the Diesel engine is capable of using the heaviest of crude oil without any difficulty, providing that it does not contain deleterious matter, such as grit, sand and vegetable substances. Its progress as a power producer has been remarkable, both for use on land and sea, and, in fact, under the sea. It is manufactured in sizes from 8 B.H.P. to 4,500 B.H.P., and experiments were carried out some time ago with an engine developing 2,000 B.H.P. in one cylinder; and after these experiments were completed, a three-cylinder engine was constructed developing 8,000 B.H.P. and running at 150 revolutions per minute.

As no particulars are available as to the results obtained from this engine, owing to its being constructed for a ship of war, I am not able to add any further information regarding it, but merely mention it here as a matter of interest indicating the high powers that may be obtained with this engine. It is usually operated on the two-stroke cycle, the four-stroke cycle, and the double-acting four-stroke cycle. The 8,000 horsepower engine referred to above was constructed on the double-acting two-stroke cycle. It is built both in a horizontal and vertical form, and the vertical engine is the one which finds most favor in both land and marine work. A brief description of the engine is given below:

The piston is of the open-trunk type, to which the connecting rod is directly coupled, as in gas-engine practice, and is of considerable length, and fitted with six to seven Ramsbottom rings; a special feature being that the combustion spaces are singularly free from pockets and recesses. An air compressor is necessary with all engines working on the Diesel principle, capable of compressing air up to a pressure of about 900 pounds per square inch, for the injection of the fuel oil into the working cylinder and for starting purposes. In the four-stroke cycle engine the cylinder cover is usually arranged with three valves—one exhaust, one atmospheric-air admission, one fuel-admission valve, and, in the case of the starting cylinders, an additional compressed-air admission valve. All the valves are opened by levers actuated by cams and are closed, and kept closed during their idle periods, by strong springs. Each valve is fitted with

a removable seat to facilitate withdrawal for cleaning and grinding. The camshaft is driven from the crankshaft at half the speed of the engine, by suitable gearing. The fuel-admission and starting valve are arranged so that both cannot be in operation at the same time. Usually, the arrangement consists of mounting the starting and fuel levers on an eccentric fulcrum, their points of contact with the cam faces being fixed at right angles to each other, and when the eccentric fulcrum is moved through a given angle, one lever is brought into contact with its cam, the other being thrown out of contact. The fuel valve is usually opened about one-quarter per cent. before top dead center, and closed after the piston has moved about ten per cent. on its down stroke.

On engines adapted for marine work all cylinders are fitted with starting valves, making it possible for the engine to be started from any position. A suitable fuel pump, which is generally driven from the cam shaft, is provided to deliver the oil-fuel valve casing. The starting of the engine is comparatively simple; the starting valves are put into action, air is admitted for a period of four to five strokes, by which time the engine will have attained sufficient momentum to permit of the starting valves being cut out, which operation at the same time brings the fuel valve into action. Oil is admitted, and the engine commences to work on fuel, gradually attaining full speed; the governor then controls the amount of fuel required to suit the load. The operation of starting takes from one-and-a-half to two minutes, and the pressure required to do so varies between 500 and 800 pounds per square inch.

In the four-stroke cycle engine air is drawn into the cylinder through a valve on the first down stroke, compressed on the up stroke, and at the point of maximum compression (about 470 pounds per square inch, and about 1,100 degrees F.) fuel oil is blown into the now highly heated air in a finely divided state by compressed air, varying in pressure from 520 to 900 pounds per square inch. Slow combustion takes place, and as the piston descends on the power stroke, fuel continues to be delivered at such a rate as to maintain the maximum pressure for about 10 per cent. of the stroke. When fuel ceases to be admitted expansion takes place for the remainder of the stroke until the exhaust valve opens and allows the exhaust gases to be expelled during the next up stroke of the piston, thus completing the cycle.

A power stroke is obtained every two revolutions. There are three types of two-cycle engines at present in use:

(1) The type in which the cylinder covers are fitted with scavenging-air admission, fuel admission, and starting valves. Exhaust valves are eliminated, exhaust ports being arranged instead in the cylinder walls, and which are covered and uncovered by the piston itself.

(2) The type in which fuel and starting valves only are fitted in the cylinder cover, the admission of the air and exit of the exhaust gases being provided for by ports arranged in cylinder walls.

(3) The type in which the fuel valve only is arranged in the cylinder cover, the admission of air and the exit of the exhaust gases being provided for by ports arranged in the cylinder walls. In this type starting is effected by admitting compressed air to the scavenge-pump piston, which pump is attached to and operates direct on to the crankshaft.

In two-cycle Diesel engines not only is an air compressor necessary for providing air for fuel injection and starting purposes, as in the four-stroke cycle engine, but a scavenge air pump is also required to provide air to assist in expelling the exhaust gases and refilling the cylinder with fresh air. There are several arrangements:

(1) In which a separate scavenge pump is arranged for each cylinder, fitted on the engine frame and operated by rocking levers and links from the main engine parts.

(2) In which a separate scavenge pump is provided per cylinder by

arranging a differential piston, in which the upper portion is the power piston and the lower portion the scavenge piston.

(3) In which a separate single pump of sufficient capacity, operated direct from the engine crankshaft, and in line with the main power cylinders.

Usually, the scavenge-air pressure required varies from four to eight pounds per square inch, according to the design of the engine. The cycle of operations for the two-stroke engine is as follows:

Assuming the piston at the bottom of the stroke, compressed air at four to eight pounds pressure per square inch is admitted through the scavenging valves or ports, and on the up stroke is compressed to 470 to 500 pounds pressure per square inch, as in the case of the four-stroke engine; fuel is admitted as previously described, and expansion takes place until the piston descends to within 20 per cent. of the bottom of its stroke. At this point the exhaust ports in the cylinder walls commence to open, and the products of combustion are allowed to escape to the atmosphere and reduce the pressure in the cylinder, lower than the pressure of the scavenging air. The scavenging ports or valves are then opened, and any remaining products of combustion are swept out of the cylinder by the incoming air, and the cylinder is refilled with clean air. A portion of the air escapes through the exhaust ports and tends to cool the bars. The scavenge valves are not closed until the piston has again closed the exhaust ports, or in the case where scavenge-air admission ports are arranged in the cylinder walls the scavenge air supply is automatically cut off by the piston covering these ports, and the cylinder is left with a full volume of clean air ready for the compression stroke.

The principal advantages of the Diesel engines are: (1) Its extreme economy in fuel consumption over a wide range of load, and the entire absence of stand-by losses. No oil is wasted in heating up vaporizers or hot bulbs. (2) Its ability to use safe, cheap, heavy, high-flashpoint residual oils obtainable over a wide range of supply. (3) Its freedom from pre-ignition due to the fact that air only is compressed, and not air and oil or vapor. (4) The entire elimination of hot bulbs or vaporizers and ignition devices. (5) Ease and safety of starting.

For starting, compressed air is used, rendering the operation simple and safe, and as no previous preparation such as preheating is required, the engine can be started from "cold" and put on load within 1½ to 2 minutes. As the introduction of the fuel is gradual, no explosion occurs, and consequently there is great smoothness in running.

Whilst the makers of the hot-bulb and vaporizer engines have increased the compression and made it possible to use light crude oils with excellent economy in consumption, continual investigations have been carried out over long periods in the chemical and mechanical laboratories of Messrs. Mirrlees, Bickerton & Day, Ltd., with almost every grade of fuel oil obtainable from all parts of the world, with the result that it has been proved that a Diesel engine can use with perfect safety and extreme economy almost every available class of heavy liquid fuel oil, with the exception of one or two oils which contain high percentages of ash; but even with these oils no difficulty is experienced in burning them, but the ash burnt off and left in the combustion spaces collects on the cylinder walls and the valves, and produces serious wear of the pistons and liners, and of the exhaust valve faces. Some years ago this firm also carried out some investigations as to the utility of tar oils, and an electric-power station abroad was fitted with "Mirrlees-Diesel" oil engines using tar oil as fuel. It was found, however, that whilst the engines would run satisfactory on this fuel when developing full and three-quarter load—that is, with the maximum heat bearing generated—trouble was experienced at lighter loads, and arrangements were made so that the engines were kept running at their maximum load by charging batteries.

Further investigations indicated that the difficulty of light-load running

could be overcome by introducing a small quantity of petroleum oil in advance of the tar oil with most satisfactory results. This is carried into effect in a very simple manner by providing a small auxiliary pump and arranging that it shall deliver the requisite quantity (*i. e.*, about 6 per cent. of the total fuel oil used at full power) of petroleum oil, or "ignition oil" as it is commonly called, to a special passage provided in the fuel-valve casing, and arranging that the delivery of ignition oil should be just before the delivery of tar oil, with the result that when the fuel valve is opened, the ignition oil passes into the heated air first, producing combustion and increasing the temperature, so that when the tar oil which follows it is delivered to the cylinder, it immediately ignites, and combustion takes place. Ignition oil is delivered in a definite quantity at all loads, so that spontaneity of combustion of the tar oil at all loads is definite.

One of the main difficulties in large-powered cylinders of the Diesel oil engine has been the trunk piston, and Messrs. Mirrlees, Bickerton & Day, Ltd., have recently modified the design of their 20-inch cylinder engine and arranged it with a crosshead and piston rod. It is interesting to note that the first Diesel engine was made in this country by the Mirrlees-Watson Co., Ltd., of Glasgow, and was a crosshead engine.

High-powered Diesel engines have shown great progress in their application for marine propulsion purposes. The following figures may be of interest:

The total B.H.P. installed up to the close of 1915 was 113,200, and the total B.H.P. on order for 1916 is 99,000. The largest power on a single shaft (two-stroke cycle engine) was 1,800 B.H.P. The largest power on a single shaft (four-stroke cycle engine) was 1,700 B.H.P. Of the above total horsepowers installed and on order, by far the largest part is comprised of engines working on the four-stroke cycle, directly reversible; that is, there are no intermediate reversing gears fitted between the engine and the propeller. This type is the one favored by Messrs. Burmeister & Wain, of Glasgow, who have turned out a large number of very successful vessels, no less than fifteen being for one owner. The largest and fastest of these ships so far built is the *Fiona*, for the East Asiatic Co., which vessel has twin engines of 2,000 I.H.P. each. These engines have each six cylinders having a bore of 740 m/m, a stroke of 1,100 m/m, and turn at 100 revolutions per minute, giving the ship a speed of about 14 knots.

The two-cycle marine engine has not, so far, been so successful that it has been largely taken up by marine engineering firms. The largest engine installed of this type is 1,800 B.H.P., mentioned above. It has six cylinders of 500 m/m bore, 1,100 m/m stroke, and turns at 100 revolutions. A still larger two-stroke cycle engine is being made by the F. I. A. T. Co., of Turin. This has six cylinders of 600 m/m bore, 900 m/m stroke, and turns at 130 revolutions per minute. A pair of these engines are intended for a depot ship now completing at Spezzia for the Brazilian Navy, and the total power of the installation will be 6,000 I.H.P.

Both two-stroke and four-stroke cycle engines are being built by most of the large marine engineering firms in this country, and a great deal of valuable experience obtained, and the author has no doubt that after the war a very considerable tonnage will be built, using Diesel oil engines for propulsion purposes. The economical advantages of this engine for vessels regularly engaged on certain trade routes are apparent. Diesel engines have also been fitted on ships of war for dynamo driving, and have also made it possible for the submarine to be operated safely, and its radius of action has been greatly increased.—"The Steamship."

FUEL OILS FROM COAL.

Mr. Harold Moore, M. Sc. Tech., recently read a paper at very short notice, on "Fuel Oils from Coal," before the Manchester Association of Engineers. He advocated the use of crude Tar as engine fuel, and also low-temperature carbonization.

In the course of an interesting paper he said that shale oil was a satisfactory substitute for petroleum, but that Scotland produced only 300,000 tons of crude oil per year, whilst the petroleum output of the United States had amounted to 33 million tons in 1913. Ordinary horizontal coal-gas retorts gave from 9 to 13 gallons of tar per ton of coal (about 5 per cent. by weight), while low-temperature carbonization yielded from 10 to 20 per cent. of tar. Mr. Moore holds that the lighter fractions of the tar distillate were known as creosote, and served both for timber preservation and as fuel for Diesel motors; this country produced annually about 70,000,000 gallons of this creosote, of which 28,000,000 gallons had been exported before the war. Tar oils from low-temperature carbonization being hardly on the market, the possibilities of raw tar as engine fuel had to be studied. Raw tars cost about 25s. or 30s. per ton now, which was half the price of the distillate; heavy tars yielded about 25 per cent. of their weight as tar oils, so that the direct utilization as fuel of raw tar, which was made all over the country, and not in special works only, would make four times the material available for power purposes. In calorific power tars were 16 per cent. lower than average petroleum oils. This consumption of tars, like that of heavy petroleum oils, in internal-combustion engines required, however, the use of an ignition oil and a special fuel pump and atomizer for that oil.

These problems, he said, had been investigated on the Continent, and Constam and Schläpper had found out that Diesel engines could be run on vertical-retort tars, on chamber-oven tars, water-gas and oil-gas tars, certain coke-oven tars, as well as on lignite tars, but not on tars from horizontal and inclined retorts. With this conclusion Mr. Moore was in agreement.

The requisites for fuel tars were: High hydrogen contents; low contents of "free carbon" (which would wear out cylinders and valves); high calorific power; moderately low viscosity; less than 2 per cent. of water (to prevent irregular running); low coking value (not over 15 per cent.); and low ash content (not exceeding 0.15 per cent.).

The Premier Tarless Fuel Company avoided decomposition of the tar in the retort by working annular retorts with outlets at both ends, under a high vacuum of 25 inches of mercury at 900 degrees or 1,000 degrees F., and obtained from slack 20 to 25 gallons of tar per ton; from poor cannel coal, 52 to 60 gallons; and from good cannel, 60 to 80 gallons. When dehydrated this tar made an excellent Diesel-engine fuel. He proposed to submit the hot gases to fractional condensation by cold in three successive stages; the first would yield pitch and free carbon, the second oils, the third the volatile benzene, toluene, etc.; the oils (carbolic, anthracene) of the second stage should, he said, give a good engine fuel, which, if too rich in naphthalene, could be preheated in tanks.—"Page's Engineering Weekly."

RADIO-TELEPHONY.

The transmission of speech without wires is a problem with which scientists and electricians have been dealing in one way and another for a good many years, though it is doubtful if anyone imagined until quite recently that radio-telephony could be efficiently established across the

Atlantic. Speech, however, has actually been transmitted between the giant station at Arlington, near Washington, and the Eiffel Tower. The achievement is all the more remarkable when it is remembered that not a word, or even a sound suggestive of a word, has ever been transmitted through an Atlantic cable, nor is there reason to think that this will be possible whilst present methods are in vogue. Working jointly, the American Telephone and Telegraph Company and the Western Electric Company are said to have obtained results little short of marvellous considering the distance involved. To American engineers the credit for this achievement belongs, but it must not be imagined that wireless experts in this country have done nothing in this particular field. As far back as 1913 Mr. Round, of the Marconi Company, carried on conversation without wires between Marconi House and Berlin. In fact, such excellent progress was being made that had it not been for the intervention of the war the Marconi transatlantic stations would in all probability now be in wireless telephonic communication. Next to nothing of a technical character concerning the tests between Arlington and Paris has been published, although there is reason to believe that the methods adopted are very similar to those employed in the latest experiments conducted by Mr. Round.

Wireless telephony is a broad subject and does not lend itself to brief treatment. The fundamental principle, however, is that a stream of radiations constant in wave length and amplitude must be emitted from the transmitting station continuously or with interruptions not fewer than several thousands per second, and upon this uniform flow of radiation changes of some kind or another must be imposed in quantitative accord with the acoustic vibrations to be transmitted. At the receiving end the apparatus must be capable of giving in a quantitative manner audible indications of these changes conveyed by the idiosyncrasies of the incoming radiations. The Poulsen arc provides a practically constant stream of waves, whilst certain forms of the quenched-spark discharger give a rapidly-interrupted stream. The sparks in this latter case must occur one after the other so rapidly that their frequency is above the audible range where the telephone and ear are not sensitive, and any resultant tone would not interfere with the reception of speech. Generally speaking, in the past the continuous high-frequency currents have been generated either by an alternator of special design or by some form of high-voltage direct-current arc shunted by a capacity and inductance. Numerous methods have been proposed for obtaining the acoustic control. If the radiations are generated by an arc the microphone may be made to modify the supply current to the arc or the length of the arc or the gas supply or magnetic field. But in spite of the wide range of possibilities in this connection the difficulties at the transmitting end are infinitely greater than at the receiving end. The matter of sufficient energy control is the one great problem in wireless telephony which has stood in the way of attaining distance ranges comparable with those accomplished in wireless telegraphy. At the receiving end the difficulties are, comparatively speaking, insignificant for any continuously and rapidly-acting quantitative detector that can be used with a telephone receiver will serve for receiving, irrespective of all arrangemnts at the sending station. From the small amount of technical information given with respect to the Washington to Paris trials no definite conclusions as to the methods adopted can be drawn. Those in a position to form an opinion, however, consider that in all probability an ordinary microphone for speaking was used, the current through it being amplified many times and then employed to modify a high-frequency current of great power produced by the new oscillating Fleming valve. If this assumption be correct, then the methods correspond very closely with those adopted by Mr. Round. Since the oscillations produced by this

method are very constant in wave length and amplitude great magnification is possible. Most recent progress seems to have been along the lines of amplifying the current and producing waves by various kinds of vacuum tubes. Mr. Round and the Marconi Company have utilized the vacuous detector with one hot and two cold electrodes for the production of continuous high-frequency oscillations, and others have carried out important work on the same lines. Much trouble was met with in the past with microphones connected in the aerial circuit. To obtain the requisite control over the outgoing waves the microphones had to be made for currents much heavier than usual, but owing to the magnifying power of Fleming and other valves these difficulties have now been removed. So far as it is possible to gather from the information available these vacuum-tube methods of producing and amplifying waves are at the present time the most hopeful. High-frequency alternator systems have been used ever since the first trials at Fessenden, nearly ten years ago.

They do not, however, appear to meet commercial requirements. Their high cost, low frequency, and consequently long wave length and difficulty of speed regulation, hinder their application. With the theory of vacuum-tube generators and amplifiers we hope to deal at some other time. Our present object has been to discuss the situation rather than to enter deeply into the theories involved. But it must not be imagined that we have by any means covered all the workable systems. During the past ten years an enormous amount of work has been done, and between the years 1913 and 1914 experimenters were particularly active. About this period several distinct methods were developed by different workers. Of these the Japanese T. Y. K. system, the American Janke, the Ditcham, and the reaction-valve systems are the most important. The first two systems are arc systems, the third a quenched-spark system, and the fourth, as its name implies, a vacuum-tube system, as developed by Marconi, Round and others.

There are two promising fields for radio-telephony. The first is for long distances where wire telephony at present is impossible over submarine cables. The other is for comparatively short distances, between ships, for instance, and from shore to ships some distance out at sea. In the latter case wireless transmission might be used in connection with land lines, so that communication might be established with the same ease as when talking from one city to another. For work of this kind neat and compact sets are made, operating on the vacuum-tube principle. It was with a set of this kind that the first tests were conducted by Mr. Round between two Italian cruisers. With ridiculously small aerial currents, such as .2 ampères, it was possible to transmit speech 70 kiloms., this high efficiency being due to the absolutely silent and constant character of the oscillations produced and to the consequent ability to magnify at the receiving end. It is doubtful if radio-telephony will ever supersede the present wire system for short distances over land, but it will undoubtedly be of immense value where the wire telephone is impractiacble. The development of radio-telephonic apparatus has been a slow and laborious process, involving much perseverance and expense. At times any real measure of success appeared hopeless, except for short ranges of transmission. But now that speech has been transmitted between Paris and Washington the outlook is altogether different, and there is every reason to believe that sooner or later radio-telephony will establish itself permanently as a means of communication throughout the world.—"The Engineer."

THE PROPERTIES OF OILS AND THEIR RELATION TO LUBRICATION.

By George B. Upton,

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The ordinary users of oils for lubrication of machinery, and many of the sellers, have very little notion of what those properties of oils are which are concerned in lubrication, and how lubrication comes about. Hence the following brief sketch of the meanings and purposes of various tests to which oils may be put.

1. *Test for acidity.*—Acids in oils, in amounts greater than traces, cause rusting and pitting of metal surfaces, and will soon put shafts or bearings out of commission. Alkalis must also be avoided. Oils should be chemically neutral, neither acid nor alkaline. All oils form acids more or less rapidly in the presence of heat and moisture; more rapidly the greater the heat. With mineral oils this action is usually negligible; with animal and vegetable oils it is considerable. Soaps used to alter the viscosity of oils also decompose by heat. Mineral oils are often bleached during manufacture by the use of sulfuric acid, which may fail of removal or neutralization in the later processes, and so remain in the oil as sold. An oil which is neutral when purchased, and which does not get very hot in use, can be expected to stay neutral.

2. *Test for ash.*—All pure oils, when heated and ignited in the open air, burn quite completely, leaving almost no ash. Mineral soaps put into the oils leave ash. The ash test therefore becomes practically a test for the presence of soaps in the oil. Whether or not these soaps are objectionable depends upon the use to which the oil is put. Soaps usually raise the viscosity of oils. Soaps are used freely in greases, and generally without objection. If the oil is to be used hot, however, the soap is liable to decompose and cause trouble by rusting of metals.

3. *Test for carbon residue.*—The oils consist principally or entirely of chemical compounds of carbon and hydrogen (hydrocarbons). About 90 per cent. by weight of the oil is carbon in the ultimate analysis. When heated very hot the hydrocarbons decompose partially, forming volatile combustible gases and heavier hydrocarbons, or carbonaceous matter, and even elementary free carbon. Such free carbon (and carbonaceous matter) is always in solution or suspension in oils to some extent, from the processes of manufacture. Heated very hot in closed vessels, without ignition, the oils partially distill unchanged, and partially decompose. When this action is carried to the end point there is left in the flask a thin deposit of "carbon residue." Animal and vegetable oils, and mineral oils loaded with rosins or soaps, leave larger "carbon residues" than pure mineral oils, for which the residue is only a fraction of one per cent.

The "carbon residue" test has practically nothing to do with the amount of "carbonizing" by the oil in the cylinders and on the pistons of internal-combustion engines, though it is often so taken. The conditions of the test are utterly different from the conditions in the engine cylinder. In the cylinder the oil is spread out in thin films on hot metal, and exposed to the burning gases of the "explosion." These gases may or may not carry an excess of oxygen to combine with the heated oil. The amount of carbon formed in the cylinder depends on the amount of oil reaching the cylinder and the conditions of exposure in the cylinder. The amount of oil depends on the mechanical fit of piston and cylinder, especially the condition of the piston rings, and on the viscosity of the hot oil on the cylinder walls. The lower the viscosity the more oil will go up past the pistons. After the oil has gone past the pistons a "lean" mixture in the combustion space will have excess oxygen to burn up the oil more or less completely, while a "rich" mixture will not have excess oxygen, the

oil cannot burn, and the rich mixture may itself be depositing lampblack carbon.

4. *Oxidation test.*—Oils gradually oxidize by contact with air at exposed surfaces of oil. The rate of oxidation increases with temperature. The product of the oxidation is resinous or gummy, and spoils the oil as a lubricant if much oxidation occurs. Mineral oils oxidize very little or not at all. Some, but not all, of the animal and vegetable oils oxidize very rapidly. An extreme case is that of linseed oil. The action commonly called "drying" of paint is really the oxidation of the linseed oil. A lubricating oil should be practically non-oxidizing.

An oxidizing oil brings in a considerable fire risk, because when spread out thin on oily waste or cloths it will often oxidize so rapidly at ordinary temperatures as to start "spontaneous combustion." This does not happen with a non-oxidizing oil. Any oily waste will burn strongly after ignition; but a lubricating oil ought not to be able to start a fire of its own accord.

5. *Volatility test.*—Oils vaporize without chemical change, just as water does. In some cases, where the oil is kept quite hot, the loss of oil by this vaporization, or volatility, may be perceptible. Usually the vaporization loss is small, and even less than leakage losses from a machine. Mineral oils are generally more volatile than animal or vegetable oils. Volatility of lubricating oils is of little practical importance.

6. *Surface tension, or capillarity.*—This, and the adhesion of oil to metal surfaces, are the forces which cause oil to spread over the metal, and to crawl into the thin spaces between bearing and shaft, or piston and cylinder, while the metal parts are at rest. Animal and vegetable oils adhere better to metal surfaces than do mineral oils. This property is especially important in steam-engine valve and cylinder lubrication, where the oil must adhere to the metal so strongly that water, and the tendency of oil to float on water, cannot remove the oil film. Mineral oils, pure, may be displaced from a steam-engine cylinder wall or valve seat by hot water. The adding of small amounts of animal or vegetable oil to the mineral steam-cylinder oil helps the oil film to adhere. The good effect is important enough to counterbalance the bad tendency of the animal or vegetable oil to form acids. Where water is absent the mineral oils stick well enough on metal surfaces. Surface tension of lubricating oils is not usually noticed in testing.

7. *Color test.*—Color of oil has no connection with lubrication. It results from the details of the manufacturing process. Mineral oils are naturally dark, or dark red, until the carbon and carbonaceous matter are removed by filtration or bleached by sulfuric acid. The removal or bleaching of the carbon and carbonaceous matter probably does little to affect the lubricating properties, though it may make the oil prettier to look at.

8. *Emulsion test.*—A very simple test to check the purity of a mineral oil is the emulsion test. Fill a clean bottle about a quarter full of clear water (preferably distilled) and another quarter full of the oil. Shake vigorously until oil and water form an emulsion. Allow to stand for 24 hours. A pure distilled mineral oil separates completely from the water, which remains clear. Acids, soaps, rosins, etc., make emulsification easier, prevent the complete separation of oil and water, and leave the water cloudy. There will often be, after the separation, a layer of leathery foam between the oil and the water. The emulsion test is a rough one; it takes a chemical analysis to work out the real reasons for the actions shown. The fact that an oil will or will not readily form an emulsion with water is often of importance in judging whether the oil will feed or not, as an oil-pump suction will not work on the foam.

9. *Flash point, and burning point or fire test.*—It has been mentioned that when heated to high temperatures oils decompose chemically, the hydrocarbons breaking up into volatile combustible gases and carbonaceous matter, or even free carbon. The flash point is the lowest temperature at

which the decomposition of the oil by heat becomes demonstrable through the formation of a combustible, slightly explosive gas mixture over the oil surface. The burning point, or fire test, is the lowest temperature at which the production of combustible gas from the oil is so rapid as to maintain combustion after ignition. The oil does not ignite itself merely because it is at or slightly above the flash point or the burn point; external ignition is necessary.

These tests originally came in because of fire risk. The flash and burn points of lubricating oils were a generation ago often as low as 200 degrees to 300 degrees F. An ordinary hot bearing was potential material for a bad fire, for an oil fire is hard to stop after it gets under way. With improvements in the manufacture of oils, the flash and burn points have been raised so that they now lie usually above 300 degrees F., and often above 400 degrees F.

As regards fire risk, the flash point of lubricating oils is of little consequence; the burn point is the more important of the two. As regards the use of the oil as a lubricant, the flash point is more important than the burn point. The flash point puts an upper limit to the temperature of use of the oil, because the chemical decomposition changes the properties of the oil, and the formation of gas in an oil film will break the film. It is quite possible, even probable, however, that the practical upper temperature limit for the use of an oil as a lubricant occurs below the flash point, from the decrease of viscosity of the oil with rising temperature. We test for flash and burn points, therefore, to catch any oil which might bring fire risk from having these points too low, and to be sure that the flash point occurs at a temperature higher than the highest temperature at which the oil is to act as a lubricant.

There are a number of places in machinery where oils have to lubricate at decidedly high temperatures. Steam engine cylinder oils must lubricate valves and pistons at the temperature of high-pressure steam, somewhere from 300 degrees to 400 degrees F.; with superheated steam even higher. Automobile-cylinder oils must lubricate the pistons in the cylinders with a temperature in the oil film of 250 degrees to 350 degrees F. In this case a further requirement is put upon the oil that that portion of it which goes up past the pistons must burn up quickly and cleanly. Hence, the flash and burn points of automobile engine oils must be neither too low nor too high, say not under 400 degrees nor over 500 degrees F. for water-cooled engines. Stationary gas engines and marine engines run colder than automobile engines, and the flash and burn points of their oils may be lower.

10. *Heat test.*—While the oil is undergoing decomposition by heat we are interested not only in the gases given off, which give the flash and burn tests, but also in the solid or semi-solid products of the decomposition. These substances, carbon or carbonaceous matter, remain in the oil. At present there is no standardized test on this line, paralleling the flash and burn test. It may be suggested that after the flash-point temperature is known another sample of the oil might be held for half an hour at the flash-point temperature, without ignition, and then be poured off into a test tube and set away for 24 hours. It should then be examined for quantity and quality of carbonaceous matter and carbon present in suspension or settled out. Some oils form almost no carbon under this action; others will show a carbon deposit of one-quarter the volume of the oil in the test tube. Probably these latter oils would be unsuitable for high-temperature use.

The accumulation of carbon and carbonaceous matter may choke oil pipes and feed holes, and the carbon may act as a mild abrasive in bearings. The viscosity of oil tends to be slightly raised by the carbon; but the simultaneous formation of light hydrocarbons in the oil counterbalances this. The accumulation of this carbon is one of the reasons why the used oil should be periodically cleaned out of the crank case of

an automobile engine. The oil which splashes up to the inside top of the pistons is spoiled on the hot metal, then drops back into the rest of the oil.

11. *Cold tests.*—As oils are cooled they increase rapidly in viscosity, that is, become stiff or sticky liquids. They may become so stiff that they will not flow at a readily measurable rate under the action of gravity. The temperature at which, in cooling, oil first gets too stiff for gravity flow is called the temperature of the pour test. Since gravity flow is very commonly relied upon to move the oil along feed pipes to bearings, cooling below the pour-test temperature stops the feeding of the lubricant. On this account the pour test is of considerable importance as a low temperature limit to the use of the oil. Full force-feed systems of supply of oil to bearings continue to work below the pour-test temperature, down to or near to that temperature where the oil becomes practically a solid substance.

Between the pour-test temperature and the solidifying temperature oils are, in appearance and lubricating action, what "greases" are at ordinary temperatures. Some "greases" are in fact merely oils with pour-test temperatures above instead of below ordinary room temperatures. Some oils, of animal or vegetable origin, freeze as water does, at one temperature, changing suddenly from a thin liquid to a rigid solid. Most oils, however, become progressively more and more stiff or solid during cooling, with pour-test temperatures somewhere between 40 degrees and 0 degrees F., and solidifying temperatures perhaps 40 degrees or 50 degrees F. lower still.

A cold test commonly made, but of no apparent practical application, is the cloud test. Many oils contain paraffin or similar waxes in solution, and in cooling throw this wax progressively out of solution as the temperature goes below a certain point. This point is that of the "cloud test," because the separation of the wax clouds the previously transparent oil. The cloud test is usually at a higher temperature than the pour test. The cloud test may be called the temperature of beginning of the partial solidification of the oil, which partial solidification may be in part responsible for the action (or lack of action) at the pour-test temperature.

In many of the present automobile engines the oil-feed piping is internal to the engine. In such an engine even the pour test of the oil becomes of little consequence. If the oil is fluid enough to let the cold engine be started at all, the heat of friction will soon lessen the viscosity of the oil, and the chances are that the oil will be feeding properly by the time the engine will be ready to change from idling to appreciable power production. If the oil-feed piping were external to the engine, not warmed up readily by the engine heat, it would behoove the user of the engine to be very careful in the winter time about the pour-test temperature of the oil.

12. *Density.*—Density of oils is probably the property most often misunderstood and misinterpreted. Density is commonly measured and quoted in the oil trade in terms of an obsolescent scale, the scale of "Beaumé light" on a hydrometer. The relation between "Beaumé numbers" and ordinary densities, or specific gravities, with water as unity, is

$$\text{density} = \frac{140}{130 + (\text{Beaumé number})}.$$

Thus Beaumé number 29.0 means a density of $\frac{140}{130 + 29} = \frac{140}{159} = 0.880$. In strict accuracy, Beaumé number 10.0 corresponds to water as 1.000 at 60 degrees F., where the real density is 0.998 on the scientific scale, which takes water as unity at its maximum density (at 4.0 degrees C.).

Density of oils has nothing to do with lubrication by the oils; lubrication does not depend in any way upon density as a specific property

of the oil. When we speak of oils as "light," "medium" or "heavy," the property concerned is viscosity. "Light" means low viscosity; "heavy" means high viscosity. These technical usages of the words "light" and "heavy" are extremely unfortunate and highly misleading. They go back historically to a time when it was thought that oils of high viscosity were necessarily also of high density. That there is no such connection between the two properties may be readily brought out by citing liquids other than oils. Three common liquids have the properties at zero degrees Centigrade: water, density 1.00, viscosity, 0.018 c. g. s. units; mercury, density 13.60, viscosity, 0.017; glycerine, density 1.27, viscosity 46.0. It is true that oils made by the same process from the same crude oil have higher densities as they have higher viscosities; but change of process with the same crude oil as base slightly dislocates this relation of density and viscosity, and change of the crude oil entirely dislocates it.

What the density does tell concerning the oil seems to be the source of the oil—what kind of crude oil was used. Thus densities above 0.90 (Beaumé numbers less than 27) indicate in this country that Texas crude oil was used as base; densities below 0.90 (Beaumé numbers greater than 27) indicate Pennsylvania crude oil as base.

Density has an important commercial application when it is used to identify oils. Let it be supposed that a certain firm has by many trials found that one brand and grade of oil is for them the best for one use, saves the most power and gives the least cost of repairs to machinery and time out for repairs. (Cost of oil itself is less important than the value of power saving and repair saving.) Before ordering more of this oil from the jobber the firm measures the density of the oil. If they find that the new shipment of supposedly the same oil has very closely the same density as the original, it is practically certain that they got what they ordered.

In the complete testing of oils density determination is necessary as an accessory to the common types (flow types) of viscometers. These instruments measure not viscosity but the ratio of viscosity to density, so that density must be known before viscosity can be computed.

13. *Viscosity.*—Viscosity is the most important property of oils as regards lubricating action. It is practically the controlling factor, so far as the oil is concerned, in the formation and maintenance of the oil film between metal parts when the parts are in motion. Viscosity is the internal friction of fluids, the force resisting the movement of one layer of oil past the next. A liquid which, like steam-engine cylinder oil or tar, is "thick" or "stiff," and does not flow readily, has high viscosity; a liquid which is "thin," flows readily, like water, has a low viscosity. If two parallel metal surfaces of area A, distance x apart, have the space between them filled with oil of viscosity η, and move past each other with the velocity V, then the viscous force F opposing the motion is $F = \frac{AV\eta}{x}$ or viscous force = (area of oil film) (velocity difference at the two surfaces of the film) (viscosity) ÷ (thickness of film). Inverting this equation $\eta = \frac{Fx}{AV}$. With the forces, distances, areas and velocities in foot-pound-second or centimeter-gram-second units respectively, this equation tells the values of absolute viscosities in the two systems of measurement. The numerical values quoted above for water, mercury, and glycerine are in c.g.s. units. The equation for viscous force is often written $F = A\eta\left(\frac{V}{x}\right)$, in which the ratio $\frac{V}{x}$ is called the "velocity gradient."

When a shaft is at rest in a bearing, with load on, the shaft and bearing are in metallic contact on the side toward the load. The shaft being smaller in diameter than the inside of the bearing, there remains a thin crescent-shaped space between shaft and bearing metal on the side away from the load. This space will fill with oil, while the shaft is at rest, by the action of surface tension or capillarity, even against the action of gravity. The relation of parts is as shown in Fig. 1.

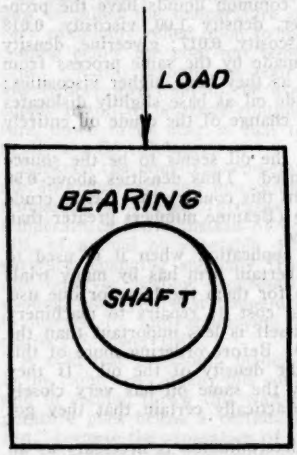


FIG. 1.—SHAFT AT REST; LOAD ON.

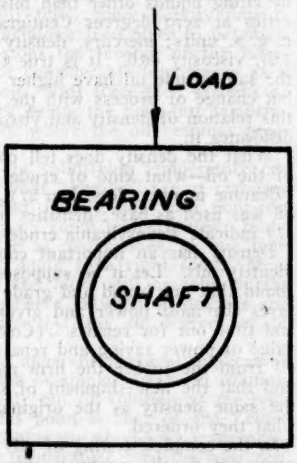


FIG. 2.—SHAFT RUNNING; OIL FILM FORMED.

When the shaft starts to turn, oil sticks to the shaft and tends to carry around with the shaft to form a film of oil between shaft and bearing on the side toward the load. The bearing, because of the load, tries to wipe this oil off the shaft, or to prevent the oil from going through. The oil layer in contact with the bearing metal stays stationary in any case. The result of the actions of the moving shaft and the stationary bearing metal is a motion of oil layer on oil layer, a motion which brings in the forces due to viscosity of the oil.

The viscous force tries to push a wedge-shaped film of oil through between shaft and bearing. The wedge is a bent one, as shown in Fig. 1. The force carrying oil into the wedge increases both with the velocity of the shaft surface and with the viscosity of the oil, or is proportional to the product of velocity by viscosity. This viscous force has against the load on the bearing the mechanical advantage of the wedge, an advantage measured by the ratio of diameter of shaft to clearance of bearing (difference of diameter of shaft and bearing). When the viscous force, or product of viscosity by velocity, becomes large enough, the oil film rather suddenly carries through, and shaft and bearing take the relative positions shown in Fig. 2. The oil film must of course always be thinner on the load side than on the unloaded side, but with good lubrication the shaft does come appreciably near to running concentric with the bearing. In other words, the thickness of the oil film toward the load, where the film is thinnest, approaches as a limit, but does not reach, one-half the clearance. There is thus a mechanical limit to the thickness of the oil film.

Because of the relationships of forces just outlined, the thickness of the oil film toward the load in any given bearing depends upon the value of the fraction $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$. The change of thickness of oil film with change of this fraction is shown in Fig. 3. The curve OCEF has film thickness for its ordinates and values of the fraction $\frac{\text{viscosity} \times \text{velocity}}{\text{load}}$ for its abscissas. OA is one-half the clearance, or the mechanical limit to the oil film thickness. The dash line HK lies at a distance BC above the base, and represents the heights of the roughnesses of the surfaces of bearing and shaft.

The course of events in lubrication during the starting up of machinery may be followed from Fig. 3. With velocity zero the fraction $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$ is zero, and there is no oil film (point O, Fig. 3). As velocity increases the fraction $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$ increases in proportion. The oil film builds slowly at first, and the bearing and shaft remain in partial metallic contact, with high friction. At some critical value of velocity the oil film becomes thicker than BC, Fig. 3, and metallic contact ceases. Thence onward to higher values of $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$ the friction of the bearing is entirely due to the oil. From C to E the oil film builds up rapidly; beyond E further change of the abscissas causes little increase of film thickness. Along CEF and beyond we are dealing with what is often called "perfect" lubrication. The term should more properly be "complete" lubrication. Between O and C lubrication is "imperfect" or better "incomplete." The formation of the oil film by viscosity and velocity action as described above is known as the "Tower effect."

The fluid friction of a bearing is given by the equation above for viscous force, $F = \frac{AV\eta}{x}$ = (area of oil film) (velocity of shaft surface) (viscosity of oil) ÷ (thickness of oil film). The form of this equation which strictly applies is $F = \eta V \int \frac{dA}{x}$, since the thickness x of the oil film varies from one element dA of area to another, both because of roughnesses of surfaces and because the shaft is not concentric with the bearing. This equation, together with the relation shown in Fig. 3 governing thickness of oil film, gives the outline of the variations of fluid friction of bearings with changes of speed, load and viscosity.

In the region from C to E, Fig. 3, thickness of oil film changes more rapidly than does the abscissa $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$. Putting this with $F = \frac{\eta VA}{x}$ shows that while the oil film thickness is rapidly increasing the fluid friction is decreasing, the factor x dominating the function. Beyond E and F, Fig. 3, the oil film has a practically constant thickness, set by the clearance of the bearing. In this region, then, since x cannot vary, increase of velocity or viscosity will increase the friction; but decrease of load will not decrease or increase the friction. Fluid friction in the ordinary bearing is not independent of load because the bearing is working in the region between C and F of Fig. 3, where change of load changes the film thickness. It is possible also that increase of load, and hence pressure in the oil film, causes an increase of viscosity of the oil; but such an effect remains yet to be demonstrated as of practical importance. The seizing of a bearing due to pressure application comes from squeezing the oil film down till metallic contact occurs.

After a machine is under way the friction causes the film and the bearings to warm up. This complicates the problem, because viscosity of oils decreases rapidly as temperature increases, especially at lower temperatures. The decrease of viscosity decreases the fraction $\left(\frac{\text{viscosity} \times \text{velocity}}{\text{load}}\right)$, velocity and load being now constant, and may affect the film thickness. The viscosity decrease may go so far as to change the film thickness from F to E to C, Fig. 3, allowing metallic contact and seizing of the bearing. If such seizing occurs repeatedly during the operation of a machine it indicates that an oil of higher viscosity is needed.

Excessive friction in machinery may occur therefore from either too low or too high a viscosity of the oil. Too low viscosity means that the oil film is not maintained. Too high viscosity, while the oil film is "perfect," gives excessive friction in the oil itself. Anyone who has cranked an automobile engine in the winter time appreciates the possibility of having too much viscosity in the oil. Too low a viscosity, with seized bearings, wrecks the machinery; too high a viscosity causes no damage to bearings, but is inefficient. The highest efficiency comes from going as near to seizing as we dare, while still playing safe.

Because viscosity varies so rapidly with temperature, and the variation is at different rates in different oils, we need in testing the oil for viscosity to find the viscosity as a function of temperature. A knowledge of the viscosity value at one temperature is not sufficient. What we need most to know is the viscosity of the oil at the temperature at which

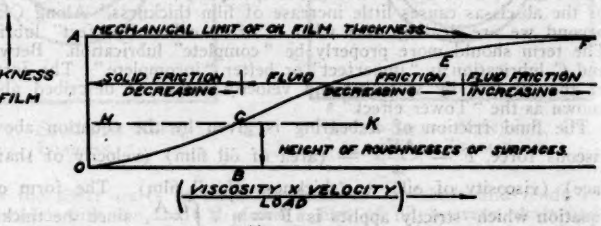


FIG. 3.

it must operate with the most severe running conditions of the machine in which the oil is used. We must be sure of having enough viscosity to maintain the oil film under the extreme conditions of operation. Then we put up with the higher viscosities when the oil is colder. Usually the range of temperature through which the oil must lubricate is narrow enough so that we can have both enough viscosity at the high temperatures and not too much at the low temperatures. In extreme cases, as the operation of an automobile engine in the winter time, we cannot get enough viscosity left at high temperatures without getting excessive viscosities at starting of the cold engine. The range of temperatures is too wide, considering the unavoidable and uncontrollably rapid change of viscosity with temperature. The ideal would be an oil which did not change viscosity with temperature.

It can readily be shown that the fluid friction of a rough-surfaced bearing is greater than that of a smooth one, other things being the same. From $F = \eta V \cdot \int \frac{d\,A}{x}$, with η and V the same in the two cases, F varies as $\int \frac{d\,A}{x}$. Let the total area in each case be 2, and the average film

thickness be 2. In the rough one let the two halves of the film area have thickness of 1 and 3; in the smooth one 2 and 2. Then $\int \frac{d\,A}{x}$ for the rough one will be $(1/1 + 1/3) = 1.33$; and for the smooth one $(1/2 + 1/2) = 1.00$. It should be said, however, that in practice the smoother bearing would probably be made with less clearance than the rougher; and in turn the oil used in the smoother bearing would be of lower viscosity. The change in the automobile engine from hand-scraped bearings to reamed bearings will probably bring about all these changes in succession.—"The Sibley Journal of Engineering."

THE TWO HUNDRED AND TWENTY HORSEPOWER RENAULT AERO ENGINE.

One of the most successful foreign aero engines of great power is the twelve-cylinder V-type Renault engine which is rated to develop 220 horsepower at a normal engine speed of 1,200 revolutions per minute. As the Renault Company of France was the first concern to produce a V-type aero engine, it is interesting to examine its latest product, which has been designed with the particular object of driving the powerful "gun-planes" of the French Air Service.

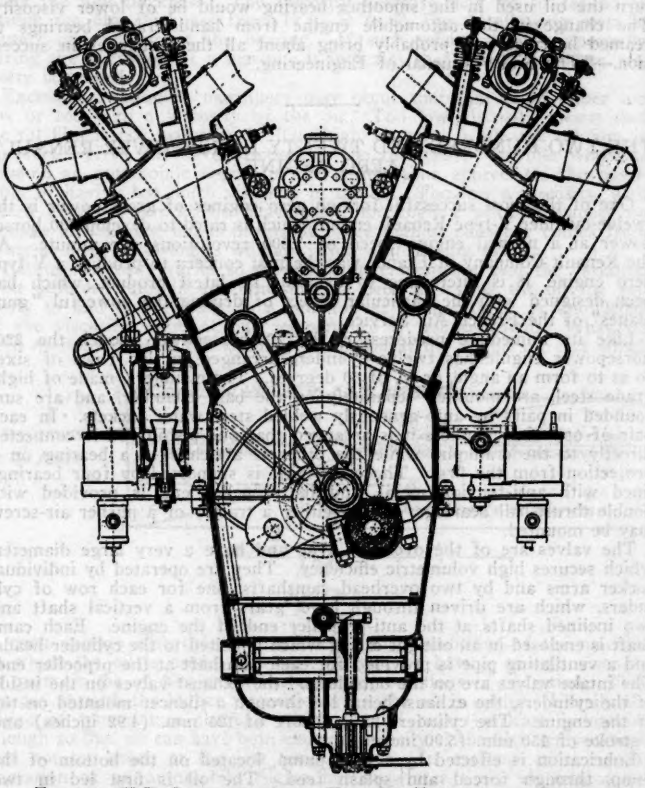
Like its immediate predecessor, the 100-horsepower model, the 220-horsepower engine has twelve cylinders arranged in two rows of sixes so as to form an angle, or V, of 60 degrees. The cylinders, made of high-grade steel, are mounted separately on the base chamber, and are surrounded in pairs by auto-geneously welded steel water jackets. In each pair of opposite cylinders it is arranged that one piston rod is connected directly to the crankpin, while the other is attached to a bearing on a projection from the first. The crankshaft is supported by four bearings lined with anti-friction metal, and its propeller end is provided with double thrust-ball bearings, so that either a tractor or a pusher air-screw may be mounted.

The valves are of the overhead type and have a very large diameter, which secures high volumetric efficiency. They are operated by individual rocker arms and by two overhead camshafts, one for each row of cylinders, which are driven through bevel gears from a vertical shaft and two inclined shafts at the anti-propeller end of the engine. Each camshaft is enclosed in an oiltight casing which is bolted to the cylinder heads, and a ventilating pipe is provided for each camshaft at the prpoeller end. The intake valves are on the outside and the exhaust valves on the inside of the cylinders, the exhaust being led through a silencer mounted on top of the engine. The cylinders have a bore of 125 mm. (4.92 inches) and a stroke of 150 mm. (5.90 inches).

Lubrication is effected by a gear pump, located on the bottom of the sump, through forced and splash feed. The oil is first led in two flows to the crankshaft bearings and to the crankpins. The cylinders are lubricated by splash. Furthermore, the oil is driven to the magneto driving gears, to the vertical and inclined shafts driving the camshafts, and finally into the camshaft casings. The wastage of oil is compensated by an auxiliary gear pump, mounted below the main pump, which draws the oil from the main tank. Both pumps are jointly operated through a worm drive by a horizontal shaft, which is driven by the crankshaft through a vertical shaft with a bevel gear.

There are two automatic carburetors, one mounted on the outer side of each row of cylinders, each carburetor feeding its individual row, through a two-branch tubing, in two sets of three cylinders. The volume of mixture admitted into the cylinders can be manually controlled by a

throttle valve. The mixture itself is slightly heated by a water jacket surrounding the carburetors and tubing, through which the water of the cooling system circulates. A proper mixture is automatically secured through a number of ports in the air chamber of the carburetor. These are fitted with annular valves whose movement is regulated by the greater or lesser force of the aspiration.



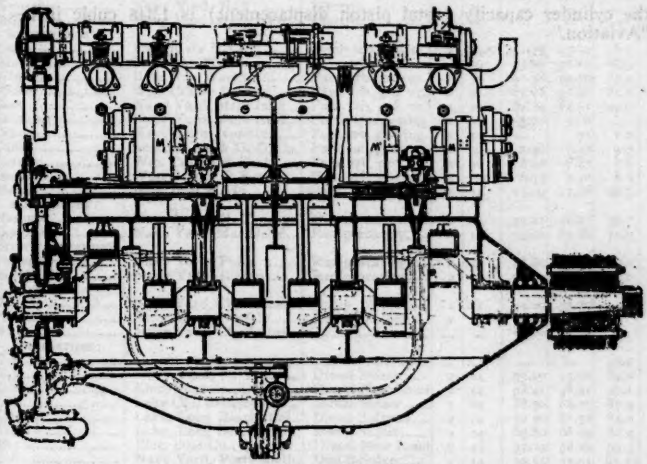
THE 220-H.P. RENAULT AERO ENGINE, TRANSVERSE SECTION THROUGH CYLINDERS.

The cooling of the engine is secured, not as on previous models through an air blower, but through a centrifugal water pump which is driven through bevel gears from a vertical shaft actuated by the engine. The water jackets of each row of cylinders are inter-connected by short lengths of rubber hose and communicate with two separate radiators through conventional inlets and outlets.

The ignition is double and is effected by four high-tension magnetos, which are driven through bevel gears by the vertical shaft actuating the camshafts. Each cylinder is fitted with two spark plugs. The magnetos

are mounted in two twin groups between the cylinder rows, each group MM′ being operated by a common gear. One magneto ignites the inner spark plugs of one cylinder row, while the other magneto ignites the outer spark plugs of the opposite cylinder row. Should, therefore, one magneto of either group break down, ignition will still be fully provided by the other magneto group.

The engine may be started either by spinning the propeller or by means of a starting crank combined with an auxiliary magneto or by a compressed-air engine. The latter follows closely the design of conventional aero engines of the radial, non-revolving type, except that it is entirely made of aluminum. The cylinders are cast integrally with the crank-case. The pistons are leather lined and their connecting rods are directly fixed to a single crank.



LONGITUDINAL SECTION OF THE 200-H.P. RENAULT AERO ENGINE.

The motor is fed from a cylinder containing sufficient compressed air for ten startings. The air reaches the cylinders by way of an air chamber through a rotating distributor which is geared to the crank and is fitted with six port holes communicating with the cylinders by means of radial channels cast *en bloc*. The rotation of the distributor effects the intake by connecting the port holes with the circular channels and the exhaust is led through the same channels back to the distributor, wherefrom another series of port holes lead to an annular chamber communicating with the atmosphere. In order to accelerate the exhaust two additional port holes are drilled on each cylinder at the lowest position of the piston stroke.

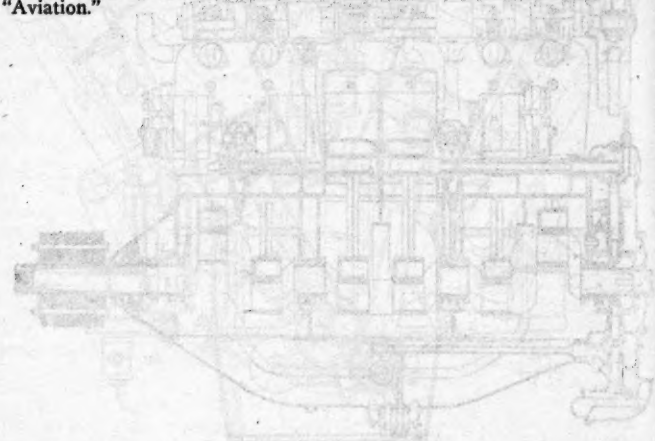
The servo-motor is connected with engine as follows: The crankshaft of the engine is coupled to a box fitted with three pawls which grip, by means of springs, a ratchet coupled to the crank of the servo-motor. When the latter is started, the pawls and ratchet are connected and the engine crankshaft will be spun till it acquires sufficient speed for insuring proper ignition. As soon as this is obtained the pilot stops the flow of compressed air and the pawls automatically disconnect themselves from the ratchet through centrifugal force.

The engine can also be started by means of a starting crank which is geared to the crank of the compressed-air motor. Because in spinning the engine crankshaft by hand, the engine speed would be too low to produce sparks of the required intensity, a fifth magneto is provided which is geared to turn ten times faster than the engine. This auxiliary magneto produces two sparks at a time, or about forty sparks for two complete revolutions of the engine. Its high-tension wire is connected with the distributors of the four main magnetos, but as the auxiliary magneto and the starting crank are coupled to the crank of the servo-motor, and therefore with the pawl-and-ratchet transmission, it will stop producing sparks as soon as the engine has started.

According to the well-known formula

$$P\ D = D^2\ 0.7854\ S\ N,$$

the cylinder capacity (total piston displacement) is 1,348 cubic inches.—"Aviation."



UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1916.		Percentage hull completed Aug. 1, 1916.	
						July 1	Aug. 1	Total.	On ship.
	BATTLESHIPS:								
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	92.54	96.75	95.4	95.2
40	*New Mexico*	Navy Yard, N. Y.	Electric	4	21	8.08	9.95	40.0	32.5
41	*Mississippi*	Newport News Co	Cur. trb. grd. cr.	4	21	42.27	46.03	50.9	42.9
42	*Idaho*	New York S. Co	Pars. trb. grd. cr.	4	21	53.05	55.71	62.3	59.0
43	*Tennessee*	Navy Yard, N. Y.	Electric	4	21	...	...	...	...
44	*California*	Navy Yard, Mare Isl'd	Electric	4	21	...	...	...	...
	DESTROYERS:								
64	*Rowan*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	94.75	97.34	96.3	96.3
65	*Davis*	Bath Iron Works	Pars. trb. grd. cr.	2	30	93.92	95.12	88.5	88.2
66	*Allen*	Bath Iron Works	Pars. trb. grd. cr.	2	30	90.58	92.50	87.4	87.0
67	*Wilkes*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	87.68	90.78	81.1	80.4
68	*Shaw*	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	62.14	64.51	43.9	44.8
69	*Caldwell*	Navy Yard, Mare Isl'd	Curtis trb. gearing	2	32	3.00	3.00	...	...
70	*Craven*	Navy Yard, Norfolk	Pars. trb. gearing	2	32	...	.71	1.6	...
71	*Gwin*	Seattle Con. & D. D. Co.	Pars. trb. gearing	2	30	2.40	2.47	9.7	...
72	*Conner*	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	6.90	7.47	8.6	3.0
73	*Stockton*	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	6.73	7.40	8.8	3.0
74	*Manley*	Bath Iron Works	Pars. trb. gearing	2	32	15.09	17.88	16.3	7.5
	FUEL SHIPS:								
14	*Maumee*	Navy Yard, N. Y.	Diesel	2	14	92.11	92.22	99.7	99.7
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	39.00	67.86	71.7	73.3
	DESTROYER TENDER:								
	Transport	Navy Yard, Phila.	Reciprocating	2	14	46.80	50.01	70.2	68.3
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	29.00	29.92	77.0	75.5
	TUGS:								
17	*Wando*	Navy Yard, Charleston	Recip. oil fuel	1	11	44.00	55.20	...	...
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	79.00	86.38	...	...
	Wave	Navy Yard, Charleston	...	...	...	73.00	77.18	...	...
	SUBMARINES:								
	G-2	...	...	...	...	...	...	92.0	92.0
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	89.5	89.3
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	99.1	99.1
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	88.70	88.75	87.4	86.5
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	70 70	81.51	83.0	80.1
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	69.80	68.09	82.4	79.7
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	97.29	98.20	99.3	99.3
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	30.73	31.51	87 53	86.35
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	97 45	97.45	...	...
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96 16	96.20	99.2	99.2
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	19.20	23.79	22.5	13.3
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	73.84	74.65	60.7	50.3
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	74.00	74.65	60.4	50.0
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	57.77	74.65	60.3	49.9
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	14.30	15.10	68.5	63.5
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	14 30	15.10	66.1	61.2
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	14.30	15.10	63.4	60.5
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	14.30	15.10	65.3	60.0
60	*O-1*	Navy Yard, Portsmouth	Diesel-New Lond	2	14	...	1.00	...	...
63	*O-2*	Navy Yard, P. S.	Diesel-New Lond	2	14	...	1.00	...	...
64	*O-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	12.4	7.6
65	*O-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	12.4	7.6
66	*O-5*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	12.4	7.6
67	*O-6*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	12.4	7.6
68	*O-7*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	12.4	7.6
69	*O-8*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	11.6	6.8
70	*O-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	11 6	6.8
71	*O-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	2.27	4.39	11.6	6.8
72	*O-11*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	3.30	3.63	28.2	22.6
73	*O-12*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	3.30	3.63	27.2	21.5
74	*O-13*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	3.30	3.63	26.1	20.6
75	*O-14*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	3.32	3.69	17 6	9.9
76	*O-15*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	3.35	3.73	17.0	9.4
77	*O-16*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	3.39	3.77	16.2	8.9

BOOK REVIEW.

THE MARINE STEAM TURBINE, By J. W. M. SOTHERN. Fourth Edition. D. VAN NOSTRAND CO., 25 Park Place, New York City. (Price $6.00 net.) Gives accurate details of many of the essential parts of marine steam turbines, together with notes on the operation of these prime movers, based upon actual experience at sea. Contains a detailed and simple description of the various uses of the temperature entropy diagram. Comprises 545 pages of text and 116 illustrations.

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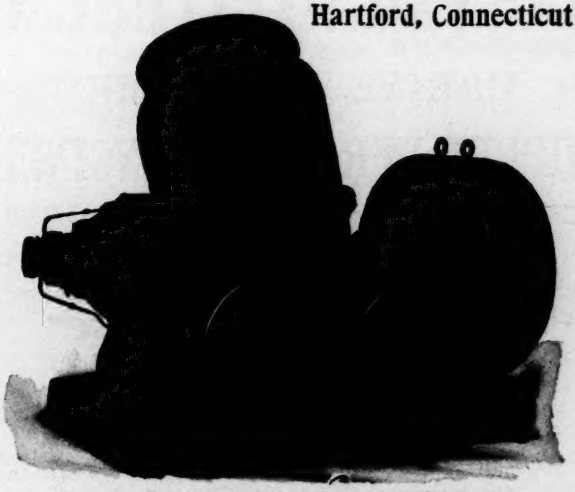

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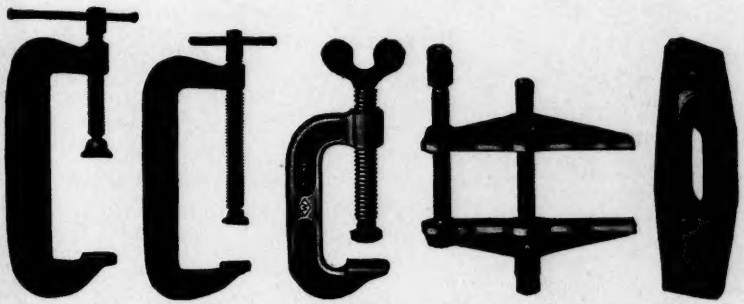

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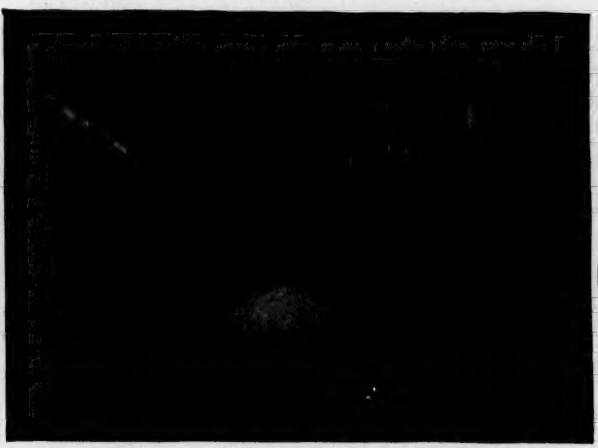

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Several of the United States Naval Vessels in commission and all those now under construction have their engines fitted with LOVEKIN PATENT ASSISTANT CYLINDERS **TO ALL OF THEIR VALVE GEARS.**

The following Shipbuilding and Engineering Companies have fitted LOVEKIN ASSISTANT CYLINDERS to valve gears on engines built by them:

New York Shipbuilding Co.	Camden, N. J.
Newport News Shipbuilding Co.	Newport News, Va.
Wm. Cramp & Sons Shipbuilding Co.	Philadelphia, Pa.
Fore River Shipbuilding Co.	Quincy, Mass.
American Shipbuilding Co.	Cleveland, Ohio.
Vulcan Works.	Stettin, Germany.
Kawasaki Dockyard Co.	Kobe, Japan.
Societa Anonima.	Milan, Italy.
N'Odero Co.	Genoa, Italy.

Many of these vessels have run over 250,000 miles without requiring any overhauling of the valve gear. These Assistant Cylinders prevent excessive wear on the eccentric straps or any parts of the valve gear **AND THEREFORE ENABLE THE VALVE GEAR TO BE KEPT IN CORRECT ADJUSTMENT FOR VERY LONG PERIODS SO THAT THE STEAM DISTRIBUTION THROUGH THE VALVES AND PORTS REMAIN AS DESIGNED.**

It is surprising to see the great consideration that is given the flat slide valve frequently used on L.P. Cylinders, few engineers forget to state that "balance cylinders will be used for the L.P. Valve Gears." How many engineers or designers have taken the trouble to find out what a poor device the ordinary balance cylinder is? It can only hold the weight of the valve gear up at one particular receiver pressure. The inertia force which is ever present and varying with the speed of the engine is always the force that causes the trouble and requires a continual expense for upkeep. **THESE INERTIA FORCES ARE PRESENT TO THE SAME EXTENT IN ALL VALVE GEARS IF THEY ARE OF THE SAME WEIGHT,** therefore, why specify the L.P. Valve Gears to be balanced and let the H.P. and I.P. Valve Gears go unbalanced when all valve gears can be balanced so easily by the LOVEKIN ASSISTANT CYLINDER.

THE LOVEKIN ASSISTANT CYLINDER performs the work of neutralizing the forces set up in the valve gear, **DUE TO BOTH THE INERTIA AND WEIGHT** of the same, and, therefore, **SAVE POWER, SAVE WEAR AND TEAR, SAVE OIL, SAVE REPAIR BILLS AND KEEP THE VALVE SETTING CORRECT FOR MUCH LONGER PERIODS THAN IS POSSIBLE WITHOUT THEM.**

The valve gear can be designed much lighter.

Balancing the inertia and gravity forces in the valve gears enables the main engine to be balanced to a far greater extent than is possible where the valve gear forces are not balanced. This was proven quite recently in the case of some high-speed destroyers in the United States Navy, where comparative tests were made with and without LOVEKIN INERTIA CYLINDERS.

NOTE.—NO FORM OF PISTON VALVE OR ORDINARY BALANCE CYLINDER CAN BE DESIGNED TO DO MORE THAN CARRY THE WEIGHT OF THE VALVE GEAR AT ONE CONSTANT PRESSURE IN THE RECEIVER. THEREFORE THE GREATEST FORCE, OR THAT DUE TO THE INERTIA OF THE VALVE GEAR, IS NEGLECTED.

Where flat slide valves are used it is necessary to provide a relief frame at the back of the valve so as to eliminate friction. I have about one thousand of these Cylinders in use at the present time, all of which are giving the greatest possible satisfaction to the owners.

I refer you to the following owners and builders, some of whom have had years of experience with my Cylinders:

American-Hawaiian Steamship Company, New York City, N. Y.	(2 Engines, 3,600 I.H.P.)
United States Navy Department, Washington, D. C.	(20 Engines, 200,000 I.H.P.)
Pacific Mail Steamship Company, California	(8 Engines, 30,000 I.H.P.)
Pacific Coast Steamship Company, Seattle, Washington	(5 Engines, 16,500 I.H.P.)
Merchants & Miners Transportation Co., Baltimore, Md.	(3 Engines, 9,600 I.H.P.)
Gulf Refining Company, New York City, N. Y.	(4 Engines, 12,000 I.H.P.)
Texas Oil Company, New York City, N. Y.	(2 Engines, 5,000 I.H.P.)
Standard Oil Company, New York City, N. Y.	(4 Engines, 7,200 I.H.P.)
Old Dominion Steamship Co., New York City, N. Y.	(1 Engine, 1,800 I.H.P.)
Coastwise Transportation Co., Boston, Mass.	(5 Engines, 9,000 I.H.P.)
Mallory Line Steamship Co., New York City, N. Y.	(1 Engine, 7,000 I.H.P.)
American Shipbuilding Co., Cleveland, Ohio	(12 Engines, 20,000 I.H.P.)
Kawasaki Dockyard Co., Kobe, Japan	(2 Engines, 8,000 I.H.P.)
N. Odero Company, Genoa, Italy	(4 Engines, 20,000 I.H.P.)
Vulcan Works, Stettin, Germany	(2 Engines, 5,000 I.H.P.)
United States Revenue Service, Washington, D. C.	(2 Engines, 3,600 I.H.P.)
Chesapeake Bay Line Steamer *Columbia*,	(1 Engine, 3,500 I.H.P.)
Miscellaneous Small Engines	(14 Engines, 15,000 I.H.P.)

TOTAL I.H.P. IN USE 376,800.

Many of the above engines have been in use for periods of from five to ten years, and I have numerous letters from the owners testifying to the great advantages of my patent Assistant Cylinders.

MR. LOVEKIN, CAMDEN, N. J., *June 28, 1913.*
Chief Engineer, New York Shipbuilding Co., Camden, N. J.

DEAR SIR:

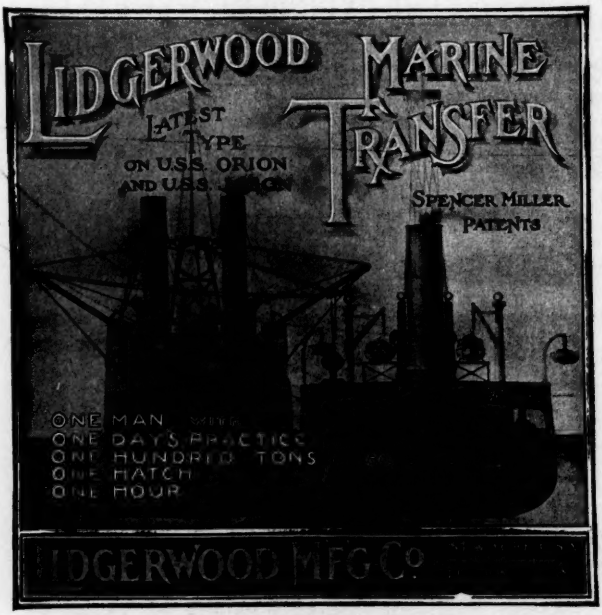
I have been Chief Engineer of the S. S. *Governor* for the past 6 years, and your Assistant Cylinders were installed on her. I found them to be a very good thing for the eccentrics, for after running 6 years the low-pressure straps had only worn $\frac{1}{16}$ of an inch, and the H.P. and M.P. the thickness of a tin shim. The engines are twin screw 5,000 I.H.P.

We ran about 370,000 miles and averaged 22 landings a month. We only had water on an eccentric once in that time, and that was when the steam pipe to the low-pressure Assistant Cylinder broke, and while we were repairing it we had to run water on it as it would not run with oil.

The only repairs we had to make to the cylinders was to put in new snap rings once a year, and I can recommend as a great saving on the valve gear and the eccentric straps.

Yours truly,

B. K. MARTLAND,
Chief-Engineer, S. S. *Congress*,
Pacific Coast Steamship Co.



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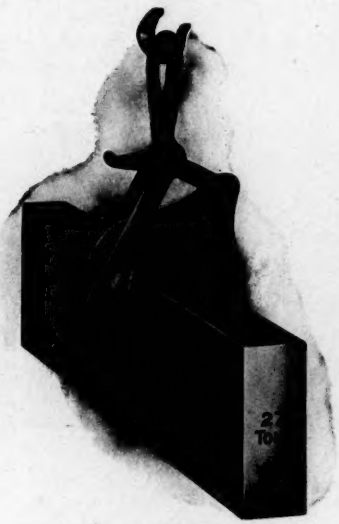

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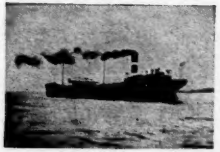

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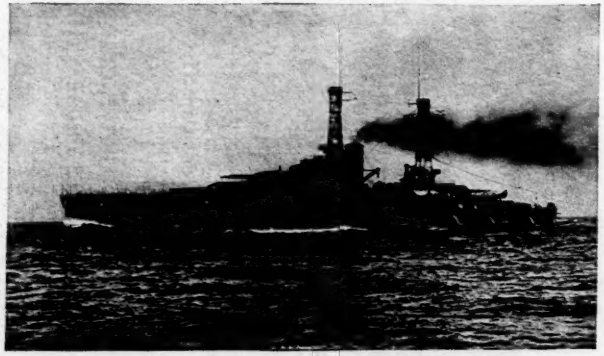

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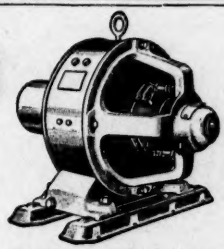

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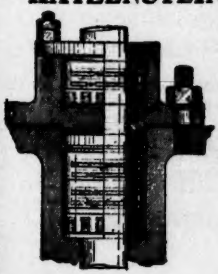

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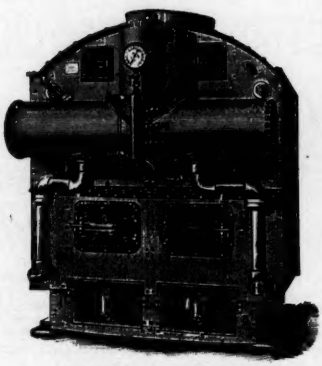

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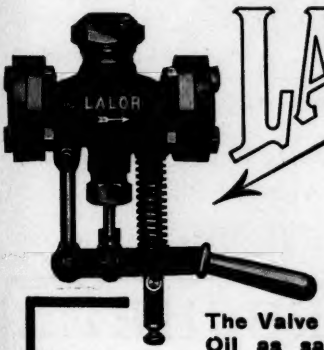

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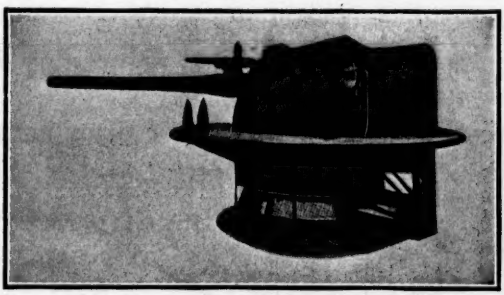

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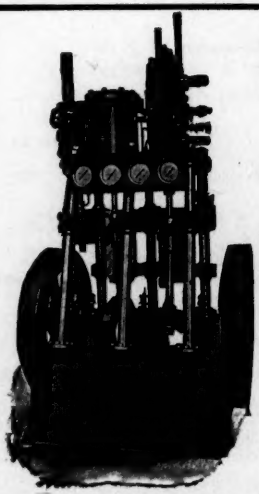

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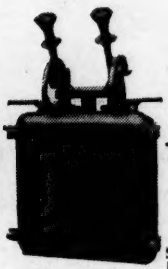

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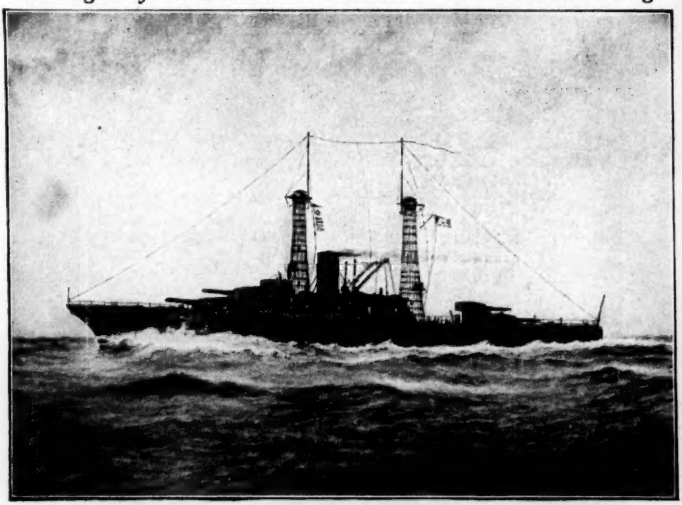

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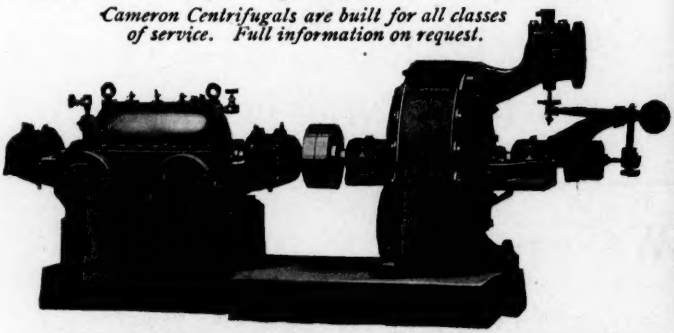

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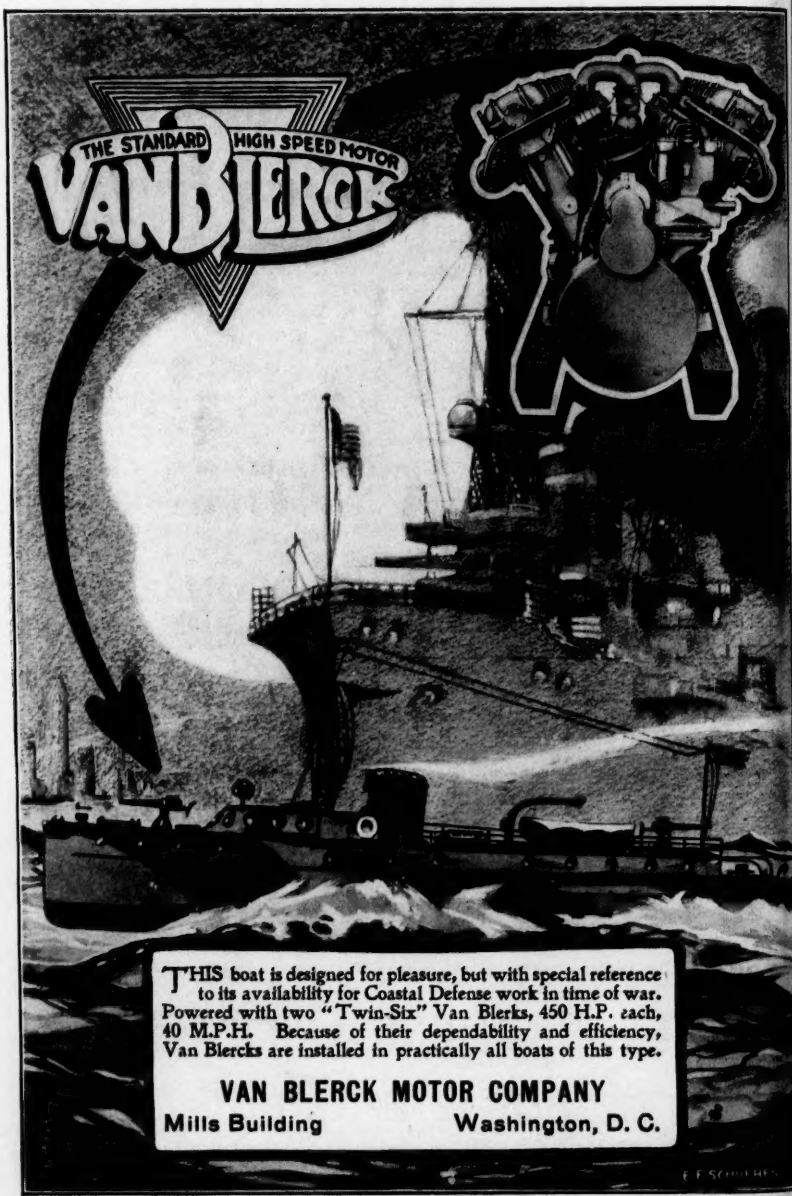
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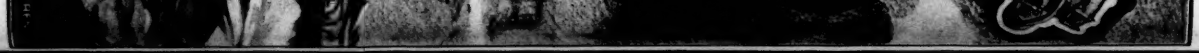

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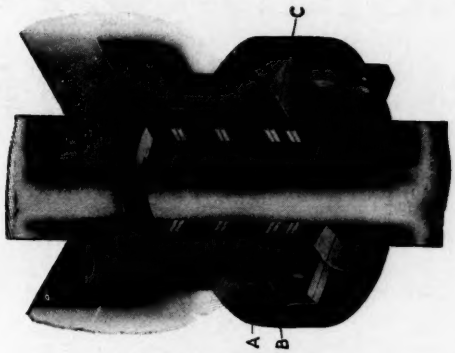

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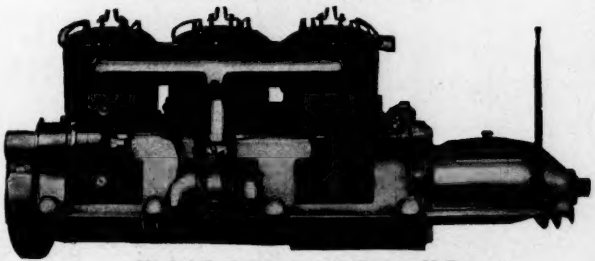

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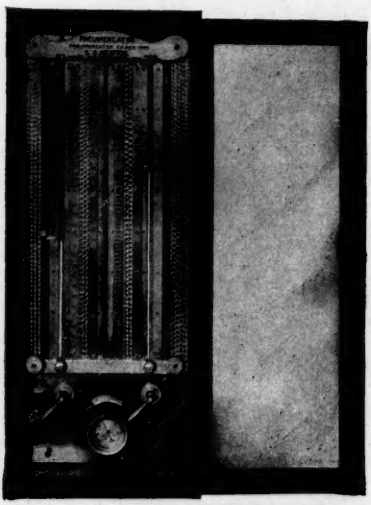

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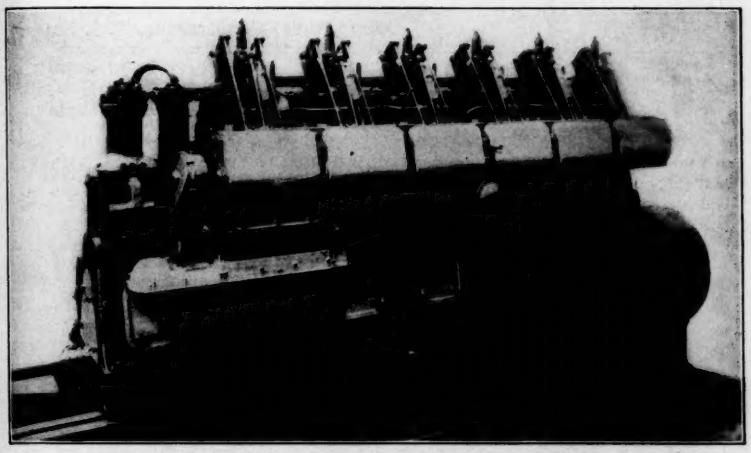

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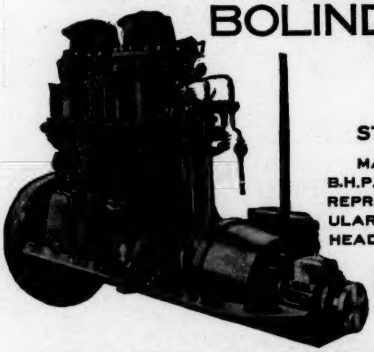

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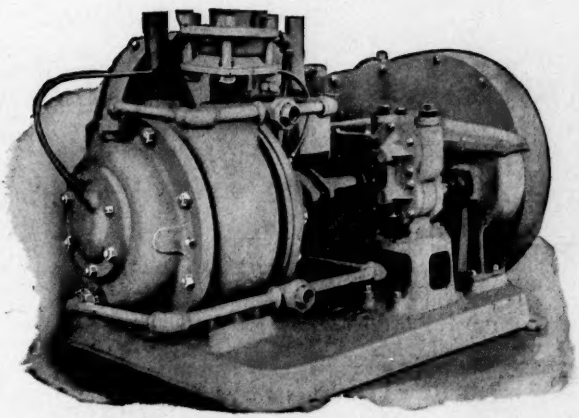

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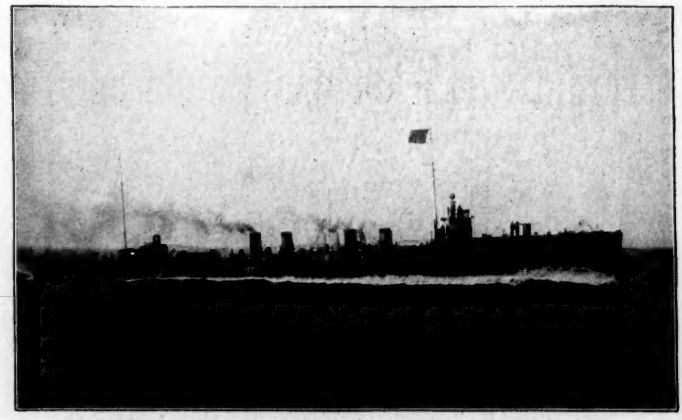

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